FOR ENERGY CONSERVATION

By

MAJOR ADYA PRASAD PANDEY

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DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

MARCH, 1989

FOR ENERGY CONSERVATION

A Thesis Submitted
In Partial Fulfilment of the Requirements
for the Degree of

MASTER OF TECHNOLOGY

105316

By

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to the

DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

MARCH, 1989

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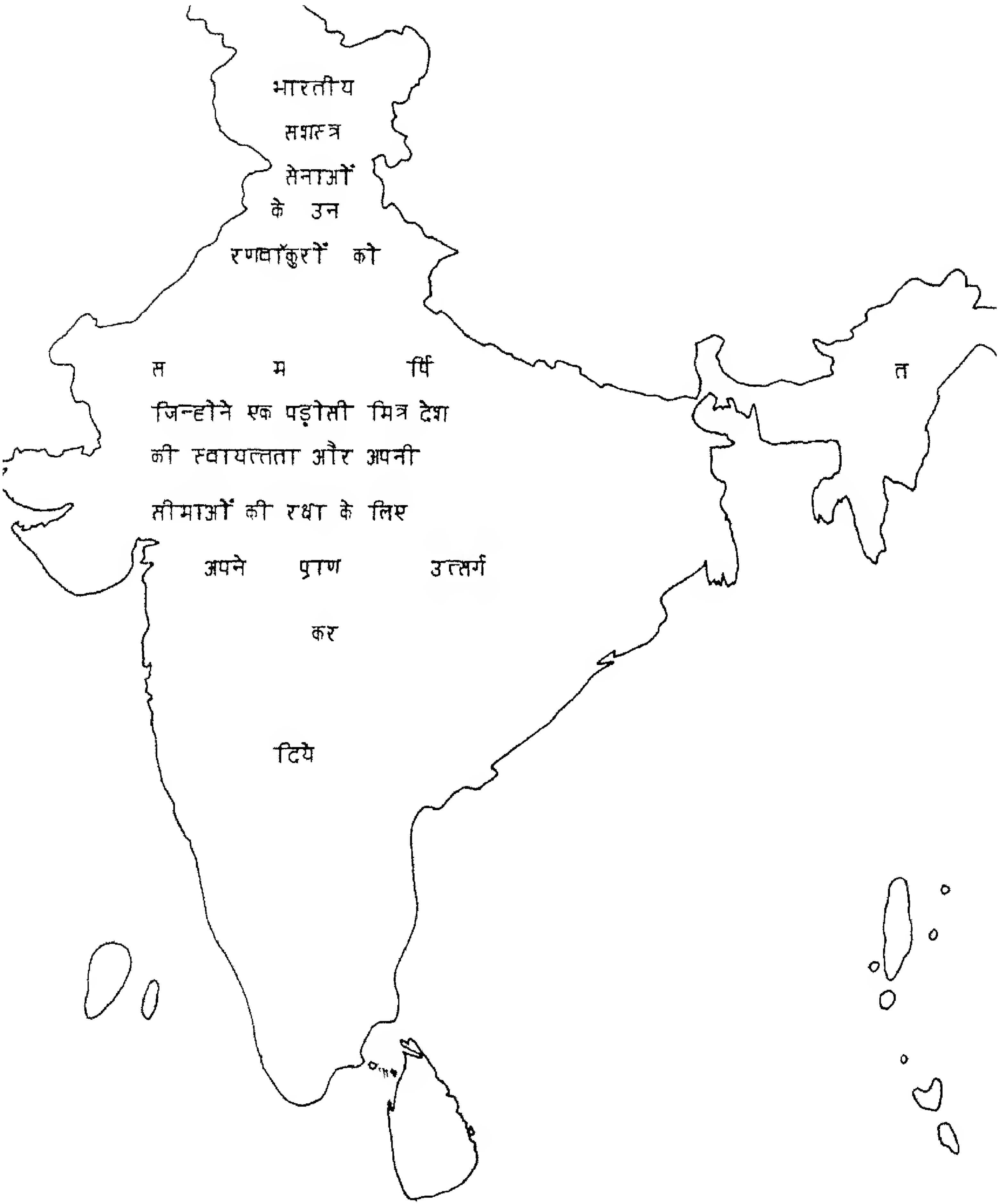

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CERTIFICATE

Certified that this work on ' Development of 1.5 Ton Hybrid Airconditioner for Energy Conservation' by Major Adya Prasad Pandey has been carried out under my supervision and that this has not been submitted elsewhere for a degree.

March, 1989

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ACKNOWLEDGEMENTS

It has been a matter of great pride for me to work under the inspiring guidance of Dr. Manohar Prasad, a nationalist in the true sense of the word. His novel ideas and deep concern for energy conservation, motivated me to accomplish this modest piece of work. It has been a privi-ledge to work under him.

I am greatful to Dr. Keshav Kant for his encouragement throughout the programme.

I am highly obeliged to Shri PN Mishra and Shri SK Mis for their help throughout this work.

I wish to extend my gratitude to Shri Bakshish Singh and his wel-knit team of Airconditioning Maintenance Unit of IIT Kanpur for their valuable help during the course of this work.

I would like to extend my sincere thanks to friends like Shri AK Jaiswal and Major VI Trivedi for their valuable assistance.

I would like to thank Shri RC Vishwakarma for his enthusiastic, neat, speedy, flawless typing and Shri BN Sriva tava and Shri SS Kushwaha for their splendid drawings.

Finally I am thankful to my wife Smt Kamla Pandey for managing the affairs at home and my children Ashwini, Seema and Saumya who acted as fast moving courriers during this wor

- A.P. Pandey

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NOMENCLATURE

A	Cross-sectional area of the structures, m^2
A_b	Body surface area, m^2
C_p	Specific heat capacity kJ/kg.C
C_{tn}	Thermal capacity of the wall kJ/kg.C
d	Declination angle, degrees
F_s	Sunlit fraction
F_{ss}	Angle factor between the surface and sky
f_g	Ratio of surface area of body with garments to the nude body
h	Enthalpy, kJ/kg
h_a	Enthalpy of air kJ/kg
h_c	Convective heat transfer coefficient for clothing, kW/m^2. C
h_i	Inside convective heat transfer coefficient of the wall, kW/m^2.C
h_{ia}	Enthalpy of inside room air kJ/kg
h_o	Outside convective heat transfer of the wall kW/m^2. C
h_{oa}	Enthalpy of outside air kJ/kg
I_{cl}	Clothing factor

I_d	Diffused radiation kW/m^2
I_{DN}	Incident direct solar radiation kW/m^2
I_R	Reflected Radiation kW/m^2
I_t	Total intensity of solar radiation kW/m^2
K	Thermal conductivity of the material kW/m.C
l	Latitude angle
m	Mass of the wall, kg
N_{ACH}	Number of air changes required per day
N_P	Number of occupants
N_G	Heat generation per individual
p_a	Water vapour pressure of ambient air, mm of H_g
p_s	Saturation pressure of pure water vapour, bar
p_v	Partial pressure of water vapour, bar
$\dot{Q}_{total}$	Total heat transfer
$\dot{Q}_e$	Evaporative heat transfer
$\dot{Q}_m$	Metabolic heat generation
q	Heat transfer through wall kW/m^2
RH	Relative Humidity
T	Temperature at any time of the day, C
T_a	Temperature of surroundings air, C

T_{db}	Dry-bulb temperature of air, C
T_{em}	Sol-air temperature
T_g	Temperature of garment, C
T_i	Inside temperature of room, C
T_{mrt}	Mean radiant temperature, C
$T_{o(t-\emptyset)}$	Outside temperature of wall at t hours, considering time lag, C
T_{wb}	Wet-bulb temperature of air, C
$T_{w,i}$	Inside temperature of wall, C
$T_{w,o}$	Outside temperature of wall, C
t	Time, hours
U	Overall heat transfer coefficient, $kW/m^2.C$
v_a	Specific volume of air, m^3/kg
V_w	Velocity of outside air, m/s
W	Specific humidity of air, gm/kg of dry air
x	Thickness of wall material, m

α	Absorptivity of the surface for solar radiation
β	Altitude angle
γ	Azimuth angle
η	Conversion factor for heat into work
ηc	Compressor efficiency
ϵ	Hemispherical emittance of the surface
θ	Incidence angle, degrees
ϕ	Relative humidity, degrees
ψ	Zenith angle, degrees
ρ	Density of wall material, kg/m^3

ABSTRACT

In the era of global energy crisis any attempt to develop a system operating on less energy consumption over the existing system is a significant contribution in the path of energy conservation. The development of the hybrid airconditioning system is an endeavour in this direction. The present system brings down the power requirement in the hot-dry climate from 2500 W of the conventional system to 300 W and in the rainy season from 2500 W to 1650 W for 1.5 ton capacity. The significant saving in energy in the summer season would prove to be a boon to tropical countries to monitor energy supply for various other developments.

Further the modified design allows the compressor to operate cool due to reduced head pressure. Also, it gives much lower sound level to the conditioned space as compared to existing design. From the costs point of view though the hybrid system is found to be a little more expensive in initial investment, its overall cost is significantly less than that of the conventional system due to drastic decrease in its running cost.

CHAPTER -1

INTRODUCTION

1.1 DESCRIPTION

Refrigeration and Airconditioning are no more considered as luxurious items but have become a part and parcel of the present era, Statistical studies conducted on group of workers showed around 30% improvement in working efficiency and reduced absenteeism by about 20% resulting in significant improvement in overall productio Similar studies carried out in the airconditioned class rooms showed an improvement in grades by 23%, learning an grasping by 50% , research capability by 38%, ability to concentrate 85%, effective use of learned skills 30% and effective use of study time 59% [1,2]. A schematic representation of the effect of environment on workabilit of men is shown in Figure 1.1 .

Airconditioning envisages the simultaneous control of temperature, humidity, cleanliness and air motion. Th conventional summer airconditioning uses a refrigeration system and a dehumidifier as against a heat pump and a humidifier for winter airconditioning. Depending on the usage, it can be sub divided into comfort (dealing with

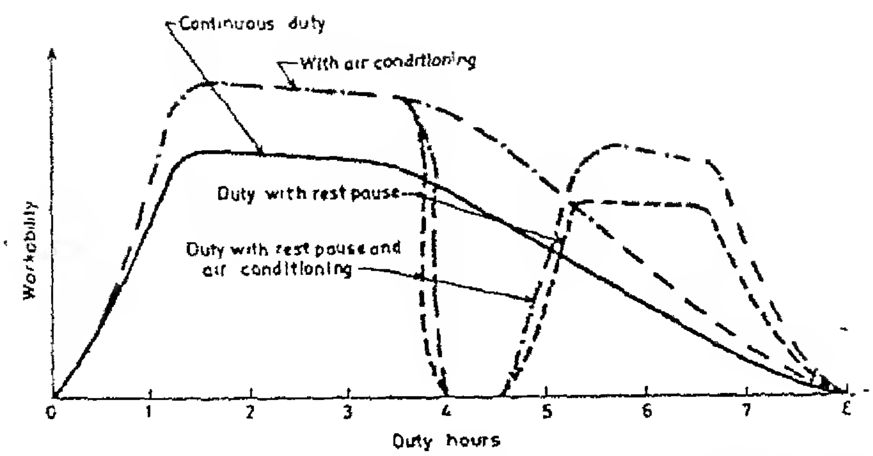
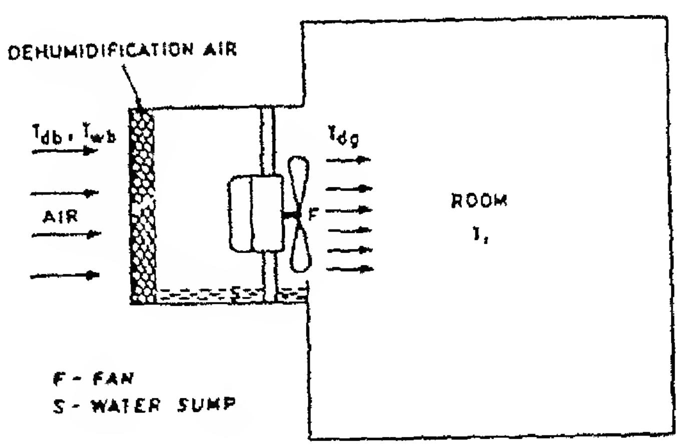
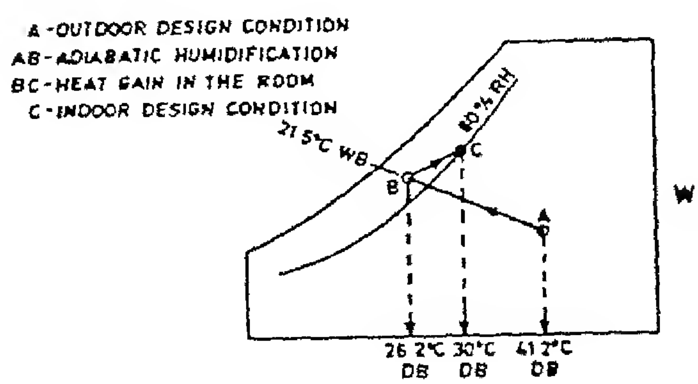


Fig 1.1 A schematic representation of the effect of environment on workability of men



(a)



(b)

Fig 1.2. (a) Evaporative cooling arrangement in room
(b) Evaporative cooling process

human comfort including noise control) and industrial (for commercial products or commodities, production shop, laboratories, manufacture of materials and precision devices, printing works, photographic products, textiles, cold storage pharmacy, computers, dams etc. In Indian context summer air-conditioning is rather more important to get better productio and to avoid health hazards. Ours being a developing country majority of its inhabitants cannot afford mechanical refrigeration gadgets for providing thermal relief in their working and living place at economical rates. In the present era of global energy crisis where fossil fuels are depleting fast, there is a need to develop cheaper and economical method for providing thermal relief.

Evaporative cooling of air provides thermal relief in hot and dry climates at much lesser cost as compared to that of vapour-compression systems. The air, by this method, is cooled by evaporation of water (sensible heat of air being taken away by latent heat of vaporization of water). The theory of evaporative air cooling is discussed in detail in Appendix A. Desert cooler is a device using this principle. Desert coolers are effective only for about three months (April to June). A typical evaporative cooling system along with the cooling process is shown in Fig. 1.2. The need of thermal comfort for rainy seasons (i.e. hot and humi climates) require search for other alternatives at economical

rates. A Hybrid Airconditioner integrating conventional airconditioner and desert cooler, capable of being operated in evaporative cooling mode and conventional airconditioner mode offers such an alternative. Details of operations of such a system are given in Sec. 1.3.

1.2 LITERATURE REVIEW

A number of extensive studies have been carried out in the field of comfort airconditioning and many investigaters [2-9] have reported their results. Some of these studies have been discussed in the subsequent paragraphs.

1.2.1 SELECTION OF INDOOR DESIGN CONDITIONS

Comfort being the primary function of airconditioning for the human beings, indoor design conditions have a lot of influence on the design of such systems. A number of investigaters have reported their studies in the recent past. Fanger [4] has elaborated the methods for getting comfort conditions using heat balance between the human body and conditioned space and developed a comfort equation after carrying out extensive field studies. The important variables influencing the thermal comfort are :

(a) activity level

(b) thermal resistance of the clothing

(c) air temperature

(d) mean radiant temperature

(e) relative air velocity and

(f) water vapour pressure

Malhotra [3] has carried out extensive field studies for determining comfort zones for the work performance of Indian workers both for hot-dry and hot-humid climates. Results of his field studies are given in Table 1.1.

Table 1.1

Comfort condition for Hot and Humid/Hot and Dry Climates [3]

	Level of comfort	Effective Temperature, ET, C	
		Hot and Humid climate	Hot and Dry climate
i.	Warm and unpleasant	27.0	26.7 - 28.3
ii.	Comfortable and pleasant (upper level)	24.5 - 25.7	24.4 - 26.6
iii.	Comfortable and pleasant (lower level)	22.0 - 22.5	21.1 - 24.3

Whitner [5] did study the effect of cost of energy for comfort and predicted the higher effective temperature of 26C in place of 22C . Evidently, the refrigeration system would take less power as compared to existing practice, [5].

Due to global energy crisis and a need for energy conservation, a compromise between energy requirement and comfort level has been reported by Ramamoorthi [6]. Inside design conditions as high as T_{db} = 30 C and ϕ = 60% having air velocity of 0.7 m/s or higher have been suggested as compared to the near static air velocity of 0.13 m/s used for the comfort environment as per the old practice (still being followed). In view of energy conservation, the ASHRAE new comfort chart shows a comfort zone having temperature as high as 26.5 C as against 21 C ET of earlier practice, Fig. 1.3.

The statistical figures obtained on the basis of opinions collected from the final year B.Tech, M.Tech. student staff and faculty members at IIT Kanpur did support the fact that inhabitants in the arid zones feel comfortable even at T_{db} = 30 C, ϕ = 60%. The choice of higher inside design condition has been arrived at as a compromise in the light of energy saving, and not as a better substitute for the conventional airconditioner [7,10].

The present thinking of elevating the comfort conditio has been supported by the latest comprehensive research carrie out by ASHRAE fellow (Kumura, et al) [9]. They have carried out extensive studies on human comfort for the hot-dry (summer season) and hot-humid (rainy season) climates

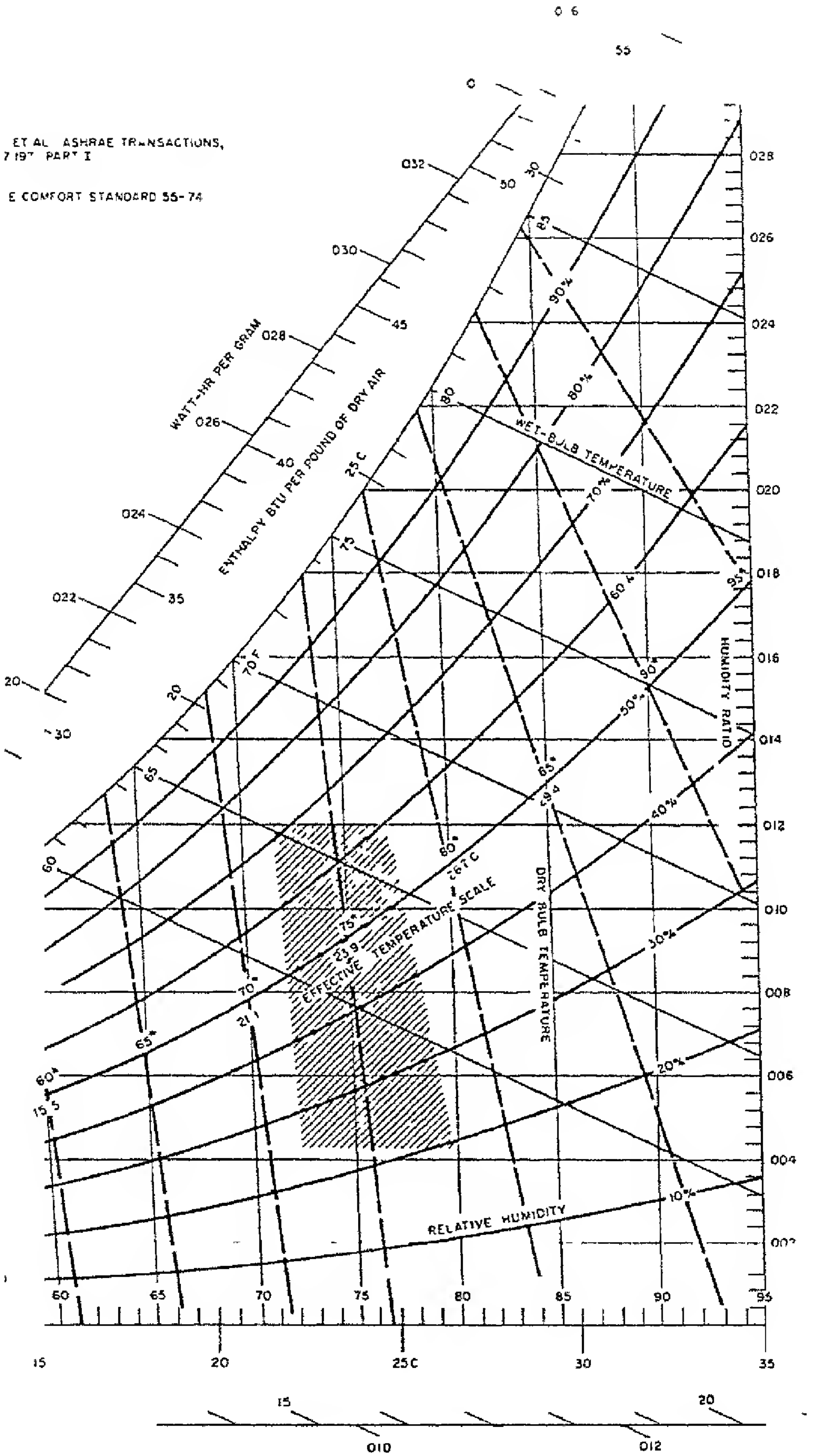


Fig. 1·3 New Effective Temperature Scale (ET*)

in the light of " Energy conservation" Their results are reported in Table 1.2.

Table 1.2

Comfort condition for Human comfort for hot-dry and hot-humid climates in the light of Energy Conservation [9]

Environmental conditions	Air motion (m/s)
27 C, ϕ = 50%	0.5
29 C, ϕ = 50%	1.2
31 C, ϕ = 50%	1.6

1.3 PRESENT STUDY

The present study on Design of Hybrid Airconditioning system was motivated by the following considerations:

(a) In hot-dry climates of our country, a desert cooler has a vast potential for providing thermal relief [10]. Hence the mechanical vapour compression system may be substituted by evaporative cooling system for the months of April to June.

(b) Fast depleting reserves of fossil fuels pose a challange to the researchers to look for systems requiring less energy in order to conserve energy for the higher priority sectors.

(c) A combination of evaporative cooler and air-conditioner (i.e. Hybrid Airconditioner) can be used for inside comfort conditions. This system is more economical than the conventional Airconditioner, used in tropical environment. Such a syste offers a good promise for energy conservation [11].

(d) It is also found that energy requirement to cause circulation of air to privide comfort is much less than that required for lowering the temperature [11

In the present study a 1.5 ton Hybrid Airconditioning system has been developed. As the head pressure of the air-cool condensers is usually 10 to 20% higher than that of water-cooled system, an evaporative-cooled condensing unit has been used. Copper tube formed is 'U' shape is embedded both sides in the wood-wool pad. Water is issued at the top and trickles down the pad whereas a blower placed inside the casing sucks the ambient air through the wetted pad.

This prototype system has provision to operate in airconditioner as well as evaporative cooling mode as per desire. While operating the system in evaporative cooling mode, the ambient air drawn through the wetted pad gets evaperatively cooled and is than supplied to the confined space for comfort purposes. In the conventional airconditioning mode condensing unit of the system works as evaperatively-cooled

condenser. Here the evaporatively-cooled ambient air cools the copper coil condenser and is discharged to the atmosphere.

The hybrid system offers a great promise for coping up with the increased power requirement for agriculture, industries and comfort airconditioning. During summer season, when the power requirement for these sectors is much above the normal, the hybrid airconditioning system spares tremendous amount of energy. Because the hybrid system need, only about 300W (in evaporative cooling mode) as against 2500W of the conventional system. Evidently 2200W can be used for the other sectors. Further, during rainy season, the hybrid system takes 1600W as against 2500W of the conventional airconditioning system. Here too there is a saving of about 900W which can be used for the above sectors. Interestingly, there is less demand of energy for agricultural sector due to natural rain. Hence, even if the hybrid system takes much more power compared to its summer value, it does not become a burden to the power management.

CHAPTER - 2

SELECTION OF COMFORT CONDITIONS FOR CONFINED SPACE

2.1 INTRODUCTION

Human beings in the present era spend greater part of their time inside a confined space. To improve their performance in order to get better output, it is desirable to provide an environment which is conducive to work. Thermal comfort plays a very dominant role in providing such an environment. By definition, " Thermal comfort is the state of mind which expresses satisfaction over the thermal environment".

Due to biological variations of age, sex, colour of skin etc. it is not possible to satisfy all occupants. But an endevour is made to create conditions conducive for work to most.

2.2 THERMAL ANALYSIS OF HUMAN BODY

A human body is the most complicated machine of nature encompassing all the processes taking place in the modern world. The body has power generating and power absorbing machines, complicated processing, automatic signalling and automatic thermal regulating devices for the body temperature etc.

The thermal analysis for the same helps understand the human comfort using energy balance between the metabolic heat generations and heat dissipation from body to surroundings.

$$\dot{Q}_m - \dot{W} = \pm \dot{Q}_s \pm \dot{Q}_c \pm \dot{Q}_e \pm \dot{Q}_r \qquad (2.1)$$

Where

$\dot{Q}_m$ is rate of metabolic heat generation inside human body

$\dot{Q}_s$ is rate of energy storage in the body

$\dot{Q}_c$ is rate of convective

$\dot{Q}_e$ is rate of evaporative } heat transfer and

$\dot{Q}_r$ is rate of radiative

$\dot{W}$ is the rate of doing work by the body.

the negative sign indicates the depletion of energy form the body.

Thermal energy for performing mechanical, chemical and other processes is met by the metabolic heat generation. Normally, the temperature of skin is kept around 36.9C and that of the inner core around 37.2C. In order to maintain the normal body temperature, a part of the metabolic heat is transferred to surroundings. As storage or depletion of energy is not required for a normal state of the human body, an analysis of heat transfer by evaporation, convection and radiation is considered for studying the comfort airconditioning.

Therefore Eq. (2.1) can be written as:

$$\dot{Q}_m - \dot{W} = \pm \dot{Q} \pm \dot{Q}_e \qquad (2.2)$$

Where $\dot{Q} = \dot{Q}_c + \dot{Q}_r$

If the ambient temperature is less than the body temperature, all the modes of heat transfer may be effective and can balance for a given air velocity. If moisture content in air increases the evaporative heat transfer decreases. To balance the net heat transfer air velocity is increased. Similarily if the dry-bulb temperature of the ambient air in the confined space is increased (but kept less than the body temperature) the larger air velocity and decreased moisture in air would balance the heat transfer.

Various combinations among the parameters T_{db}, T_{wb} and air velocity can be obtained which would give the same level of comfort. It is possible to express heat transfer from body in terms of air velocity, T_{db} and T_{wb} (or humidity ratios) in addition to the body temperature. If the ambient condition is not conducive people would experience heat stress or thermal strain. This suggests the requirement of a balance between heat generation and heat transfer to the environment without causing heat stress, for maintaining thermal comfort, $\dot{Q}_c$, $\dot{Q}_r$ and $\dot{Q}_e$ can be expressed as [1].

$$\dot{Q}_c = 6.9\, V^{0.5} (T_b - T) \qquad (2.3)$$

$$\dot{Q}_r = 42\, (T_b - T_s) \qquad (2.4)$$

$$\dot{Q}_e = 20{,}000\, V^{0.4} (w_{sb} \quad w) \qquad (2.5)$$

where T_b = Body surface temperature (K)

T = Air temperature (K)

T_s = Surrounding surface temperature (K)

V = Velocity of air in (m/min)

w_{sb} = Specific humidity corresponding to body temperature (saturated)

w = Air specific humidity

2.3 FACTORS INFLUENCING COMFORT

Temperature, humidity, air movement, purity of air and noise have predominant affect on human comfort. Colour of the surrounding also affects the comfort. Cream, light green and light blue colours are more soothing than red and other bright colours. Other factors like clothing, economical status of persons, climate to which a person is accustomed to etc ; also have considerable influence on human comfort. Temperature and humidity can be controlled by means of evaporative cooling/ appropriate airconditioning equiptment . The air movement is maintained at the desirable velocity of about 0.13 m/s (old practice) and now upto 1.6 m/s using appropriate distribution system. Purity of air is another factor having profound affec on human comfort and this can be met by adopting proper ventilation. Controlled experiments have shown that the main cause of discomfort in a closed confinement are high levels of temperature and humidity as well as odour, the reduced level o

oxygen and increased level of carbon-di oxide. However no serious discomfort arises even when the oxygen level falls to 16% and that of CO_2 rises to about 2% by volume as against 21% and 0.04% normally found in the standard atmosphere [1].

Ventilation aims at providing sufficient amount of oxygen and at the same time not allowing the level of carbon di-oxide to be too high. It has been estimated that a sedenta adult requires about 12.5 m^3 of air and releases about 0.4m^3 o CO_2 per day. At atmospheric pressure oxygen concentrations of less than 12% and carbon-di-oxide concentrations greater than 5% are dangerous even for short periods. If the environment is continuously contaminated or vitiated by the release of undesirable gases as in the viscinity of power houses, process and chemical industries it becomes essential to maintain the concentration of contaminants below certain safe levels as laid down by appropriate codes.

Purity of air is required to prevent any infection and other effects on human beings. Air contaminants may be in sol liquid or gaseous form. Solid impurities may be dusts (600 μm to higher sizes) and carbonaceous matter from chimneys (0.1 to 13 μm) etc. liquid impurities may be due to 'smog' (an air mixture of smoke articles, mists and fog droplets of such concentrations as to impair visibility and being irritating or harmful). The gaseous impurities are due to vapour or undesirable gases present in air. Air is cleaned by passing it thro

air washers and filters. Particulate impurities are removed by employing mechanical filters, inertia separaters or electro static separators (up to 0.3 μm). Air washers clean the air during its passage through the spray of water. Odours due to presence of organic matters are removed using activated charcoal whereas bacterisides like ozone and ultra-voilet radiations are used to kill bacteria in order to get sterile conditions especially in hospitals. This indicates that the comfor conditions for a hospital should be maintained with low relati humidity.

2.4 EFFECTIVE TEMPERATURE SCALE (ET) AND COMFORT HEALTH INDEX (CHI)

Some of the environmental factors affecting comfort like dry-bulb and wet-bulb temperatures and air movement can be clubbed together to yield equal sensations of warmth or cold. This index is called 'Effective Temperature'* and can be used for comparative study of various systems. Various combinations of these factors can result in the same effective temperature, indicating same level of comfort. Special charts are available for expressing the readings of dry-bulb and wet-bulb temperature and air movement into the Effective Temperatu

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* It is a **sensory** index (for expressing human comfort) that combines the effects of temperature, humidity and air moveme in a noise free pure air environment, into a single factor.

scale. This temperature is equivalent to the fully saturated air at a particular dry-bulb temperature with stagnant air (the air motion upto 0.13 m/s is regarded as stagnant).

The new 'Effective Temperature'* scale by ASHRAE is defined as the dry-bulb temperature of a uniform enclosure of 50% RH in which humans would have the same net heat exchange by radiation convection and evaporation as they would in the varying humidities of the test environment. For the ASHRAE scale (based on physiological considerations clothing is standardized at 0.6 clo, air movement still at 0.2 m/s, time of exposure 1 hour and the chosen activity as sedentary ($\simeq$1 met) [2].

Comfort-Health Index (CHI) is based on physiological and health responses and is expressed as a dry-bulb temperature of 50% RH. The CHI and associated sensory physiological, psychological and health factors are shown in Fig. 2.1 [12].

2.5 COMFORT ZONES FOR CLIMATES PREVALENT IN INDIA

Ours being a large country, there is a noticeable change in the environmental conditions encountered in differe[nt] parts and different seasons. In summer season, the climate varies from hot and dry in the central part to hot and humid in the coastal regions. Extensive field studies have been carr[ied] out by many investigators and Malhotra [3] has reported the res[ults] of his findigns for comfort zones for Indian people.

NEW ET* SCALE C	SENSATION Temperature	SENSATION Comfort	PHYSIOLOGY	HEALTH	OLD ET SCALE C
	Limited	Tolerance	Body heating failue of regulation	Circulatory collapse	
40					
	Very hot	Very uncomfortable	Increasing stress caused by sweating and blood flow	Increasing danger of of strokes, cardio vascular embarrassement	30
35	Hot				
	Warm	Unfomfortable			
30	Slightly Warm		Normal regulation by sweating and vascular change		25
25	Neutral	Comfortable	Regulation by vascular change	Normal health	
	Slightly cool				20
20	Cool	Slightly uncomfortable	Increasing dry heat loss Urge for more clothing or exercise (Behavioral reg.)		15
15	Cold		Vasoconstriction in hand and feet,	Increasing complaint from dry mucosa and skin	

conditions of ambient air for different cities in India from April to October have been plotted on psychrometric chart in Fig. 2.2(a) - (b) [8,13] . On comparison with the results found by Mishra [8] for various indoor design conditions and structure load (with and without time lag), we find that only cooling and humidification is required for most of the months for summer airconditioning. This calls for the use of a humidifier to achieve the desired inside conditions. On the other hand, in the rainy season, there is a need for cooling and dehumidification in order to maintain the same inside design condition. It implies that the airconditioner should comprise a humidifier for summer airconditioning and dehumidifier for airconditioning in the rainy season. This led to the concept of the development of a hybrid system to cope with the above dual requirements, and hence the present prototype sys of 1.5 ton capacity.

2.6 DEVELOPMENT OF COMFORT EQUATION

As already mentioned in Sec 2.2 , human comfort is influenced by the metabolic rate of heat generation within the body and the rate of heat dissipation to the surroundings. A complex regulating mechanism in the human body keeps the temperature of surface tissues or skin at about 36.9C and for deep tissues or core to about 37.2C.

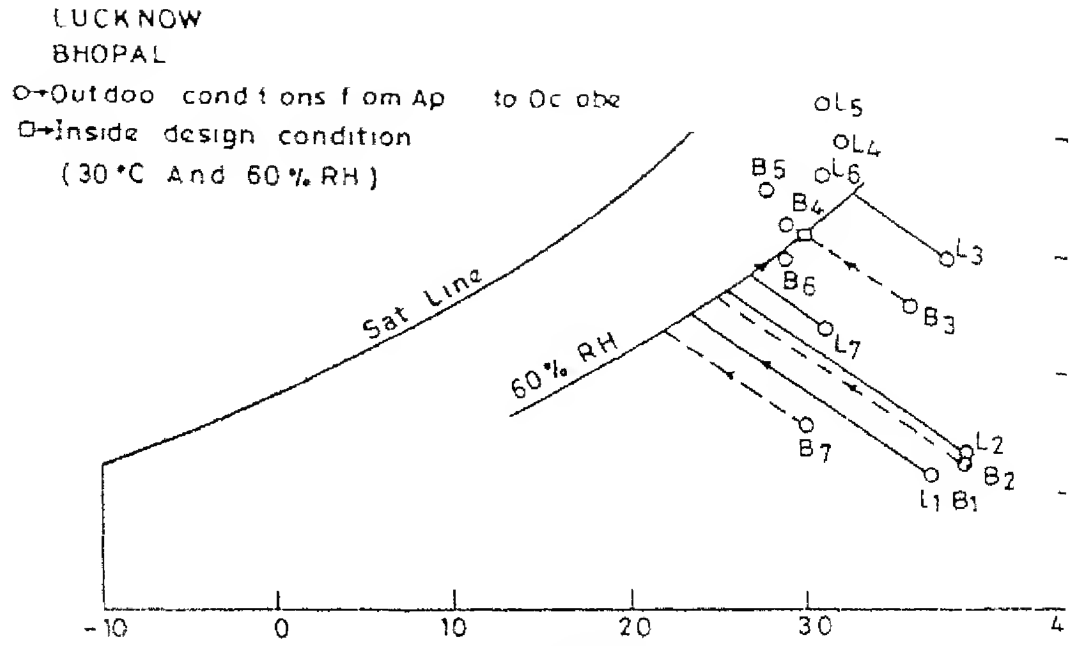


FIG. 2.2 (a)

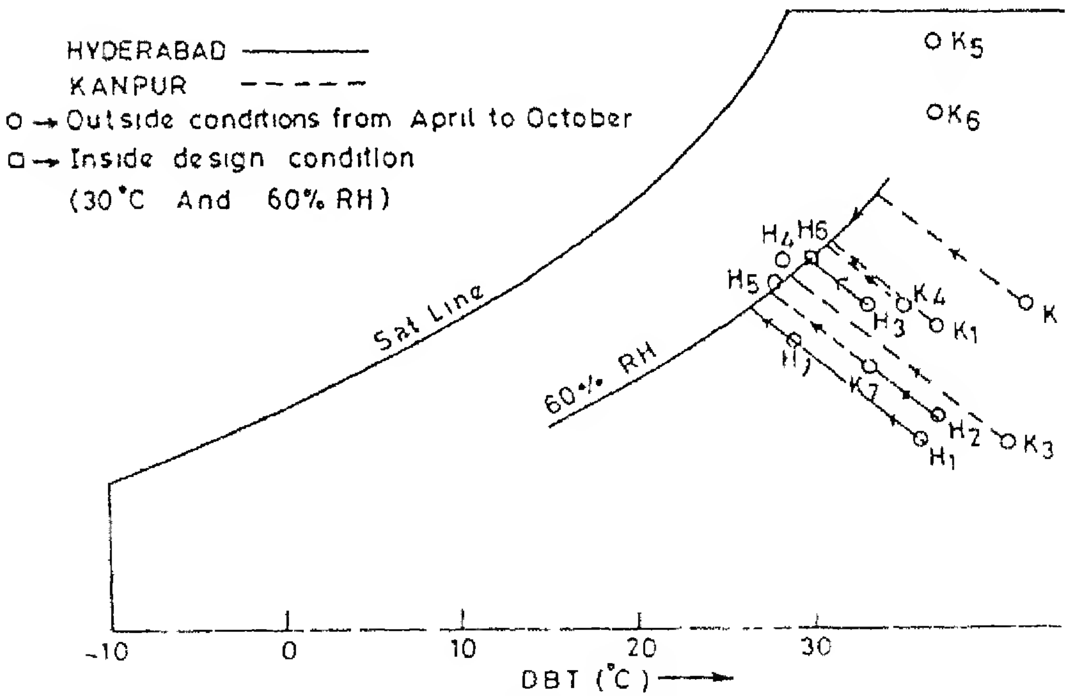


FIG. 2.2(b)

The existence of thermal balance between the body and environment is one of the necessary conditions for thermal comfort for a person under long exposure to a given surrounding condition. This heat balance is influenced by a number of factors. Considering only the main variables, an equation of the following form is reported [4]

$$f\left(\frac{\dot{Q}_m}{A_b},\ I_{cl},\ T_a,\ T_{mrt},\ p_a,\ V,\ T_s,\ \frac{\dot{Q}_e}{A_b}\right) = 0 \qquad (2.6)$$

Where

$\frac{\dot{Q}_m}{A_b}$	=	internal heat production per unit body surface area (A_b = D_u Bios area), (W/m^2)
I_{cl}	=	thermal resistance of the clothing, clo
T_a	=	ambient air temperature, C
T_{mrt}	=	mean radiant temperature, C
p_a	=	water vapour pressure in ambient air, mm of H_g
V	=	relative air velocity, m/s
T_s	=	mean skin temperature , C
$\frac{\dot{Q}_e}{A_b}$	=	heat loss per unit body surface area by evaporation of sweat secretion

Thermoregulatory system in human body adopts itself to a new set of environmental conditions within a wide range of environmental variables and thus create thermal balance, even if comfort does not exist. Hence the heat balance is not the sufficient criteria for thermal comfort.

2.7 PSYCHOLOGICAL AND PHYSIOLOGICAL VARIABLES INFLUENCING THERMAL COMFORT

Psychological and physiological both factors influence the human comfort. Heat stress Index (HSI) is a rational index indicating psychological criteria. It is the ratio of total evaporative heat loss ($\dot{Q}_e$) for thermal equilibrium (i.e., sum of metabolic + dry heat load to maximum evaporation $\dot{Q}_e$, max) to the environment itself, and this criteria is based on

(a) body temperature should not exceed 308 K (=35C)

(b) the limit of body fluid loss, i.e. the sweating rate should not exceed one litre per hour (25 kJ/h) [1] . Therefore

$$HSI = 100\ Q_e \ / \ (Q_{e\ max} \leq 2500) \tag{2.7}$$

Where $Q_e \ = \dot{Q}_m - \dot{Q}_c - \dot{Q}_r$

$$= \dot{Q}_m - 6.9\ V^{0.5}\ (308 - T) - 42\ (308 - T) \tag{2.8}$$

and $$\dot{Q}_e,\ max = 20{,}000\ V^{0.4}\ (0.0367 - W) \tag{2.9}$$

Equations (2.8) reveal that $\dot{Q}_e$ is evaporative cooling required to balance the metabolic heat generation, convective and radiative heat transfer from body where the average body temperature is 308 k, $\dot{Q}_e$, max is the required evaporative cooling when body is wet at 308 k .

Physiologically, the level of thermal comfort i.e., the sensation of comfort has been related in terms of two variables the skin temperature T_s and sweat secretion $\dot{Q}_e$ for a given activity level. The empirical relations for mean values of T_s and $\dot{Q}_e$ are given as follows

$$T_s = 308.7 - 0.0275 \frac{\dot{Q}_m}{A_b} (1-\eta) \; W \qquad (2.10)$$

$$\dot{Q}_e = 0.42 \, A_b \left[\frac{\dot{Q}_m (1-\eta)}{A_b} - 58 \right] W \qquad (2.11)$$

2.8 COMFORT EQUATIONS

Fanger's comfort equation generalizes the physiological basis of comfort so that comfort for any activity can be predicted analytically in terms of environmental parameters. It is given by

$$\frac{\dot{Q}_m}{A_b} (1-\eta) - 305.4 \, (0.00256 \, T_s - 0.7325 - p_a) - 0.42 \left[\frac{\dot{Q}_m}{A_b} (1-\eta) - 58 \right] - 1.726 \frac{\dot{Q}_m}{A_b} \left[(0.0586 - p_a) + 0.00081 \, (307 - T_a) \right] = f_g \left[h_c \, (T_g - T_a) - 3.91 \times 10^{8} \, (T_g^{\,4} - T_{mrt}^{\,4}) \right] \qquad (2.12)$$

Where T_c $\quad 308.7 - 0.0275\ \frac{\dot{Q}_m}{A_b}(1 - \eta) - 0.155\ I_{cl}\Big[\frac{\dot{Q}_m}{A_b}(1\ \eta)$

$$- 305.4 \times (0.00256\ T_s - 0.7325 - p_a) - 0.42$$

$$\{ \frac{\dot{Q}_m(1-\eta)}{A_b} - 58 \} - 1.726\ \frac{\dot{Q}_m}{A_b}\{ 0.0586 - p_a)$$

$$+ 0.00081\ (\ 307 - T_a)\ \}\Big] \tag{2.13}$$

Fanger's comfort equation being thermal energy balance between the body and surroundings is used to predict room condition for a type of activity [11] . The Newton-Raphson method may be used to get the comfort conditions for various ranges of parameters [6].

Table 2.1

Ranges of Parameters and Comfort Conditions [6]

i.	Relative air velocity	:	0.2 to 1.3 m/s
ii.	f_{cl} values	:	1.0, 1.1, 1.15 and 1.2
iii.	Clothing values i.e., I_{cl} for	:	0.0, 0.5, 1.0 and 1.5
iv.	Level of activity $\frac{\dot{Q}_m}{A_b}$	:	58, 116 and 175 W/m^2

The computed values for these parameters are reported in Appendix A of [6] .

CHAPTER - 3

ESTIMATION OF COOLING LOAD

3.1 FACTORS CONSIDERED IN COOLING LOAD CALCULATION

Cooling load for a confined space is the rate at which heat is removed from the space to maintain room air temperature at a constant value. The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. The important parameters involved in the evaluation of cooling load estimation are as follows:

(a) Orientation, geographical location and geometric dimensions of the building

(b) Properties of the building materials used

(c) Standard of ventilation required

(d) Infiltration of air

(e) Activity level and number of occupants

(f) Numbers of electrical and other heating appliances and their power rating

(g) Environmental parameters like temperature of the outside air, relative humidity and air velocity.

(h) Inside design conditions, (T_{db}, T_{wb} or ϕ, air velocity)

(i) Direct and diffused solar radiations etc.

Three methods have been developed to account for the external conditions while the inside space is kept under constant temperature, humidity and air movement. These are:

(a) Cooling Load Temperature Difference (CLTD) method also called Transfer Function Method [2]

(b) Finite difference method

(c) Sol-air temperature method [14]

Sol-air temperature method is widely used due to its simple algorithm.

3.2 DIFFERENT SOURCES OF HEAT

The cooling load for air conditioning systems for summer arises from the following:

(a) Energy transfer from surroundings to the system. These are:

i. Heat transmission through walls, doors, ceiling etc., and caused by the temperature difference existing on the two sides.

ii. Solar heat gain

(aa) Absorbed by walls or roofs exposed to solar radiation and transferred to the inside space.

(ab) Transmitted directly through glass and absorbed by the inside surfaces and furnishings.

iii. Heat and moisture carried alongwith the air infiltrating into the confined space.

iv. Heat and moisture gain **with** ventilation and infilteration air.

(b) Heat generation within the confined space.

i. Sensible and latent heat released from occupants.

ii. Heat load from electrical appliances like fans, lights, motors etc.

(c) Miscellaneous heat sources

3.3 NEED FOR HOURLY COOLING LOAD CALCULATIONS

The ambient air temperature varies in a 24 hour cycle and hence there is a variation in the heat gain accordingly. This **variation** is mainly caused due to change in solar

intensity falling on the earth surface and atmospheric conditions. The walls of the structure act like a thermal flywheel, thus storing a part of energy passing through it and transmitting it to inner/outer space some time later. Hence calculations neglecting thermal capacity of walls and based on instantaneous transmissions through structure are quite approximate and are erroneous. Also, the traditional methods of evaluating cooling load which involve assumptions like the load from each component being constant in a day, maximum value of cooling load being equal to the sum of individual maxima, are not correct. Because of the variation caused due to these assumptions, reasonably large safety margins were being provided thus resulting in the installation of a highly oversized and uneconomical airconditioning system. Since the occurance of the overall maximum cooling load at a particular time is not the summation of maxima of load due to various components, it is desirable to go in for hourly cooling load calculations to facilitate the economic selection of a system.

3.4 VARIATION IN OUTDOOR TEMPERATURE

It has been established from extensive studies carried out by various investigators that, in a 24 hour cycle, the minimum temperature normally occurs one or two hours before

sunrise while the maximum occurs about two to four hours after the solar noon [6, 8, 14]. As temperature-time history is not available for most of the places, an approximate method for evaluating maximum and minimum temperatures is adopted. This method would enable more accurate prediction of cooling load for most of places for which only maximum and minimum temperatures are available in literature [2,13,15]. The temperature variation with time is expressed in the following form:

$$T = a + b \cos(15t - c) \tag{3.1}$$

Where,

t = time in hours

T = temperature at any time t hours, C

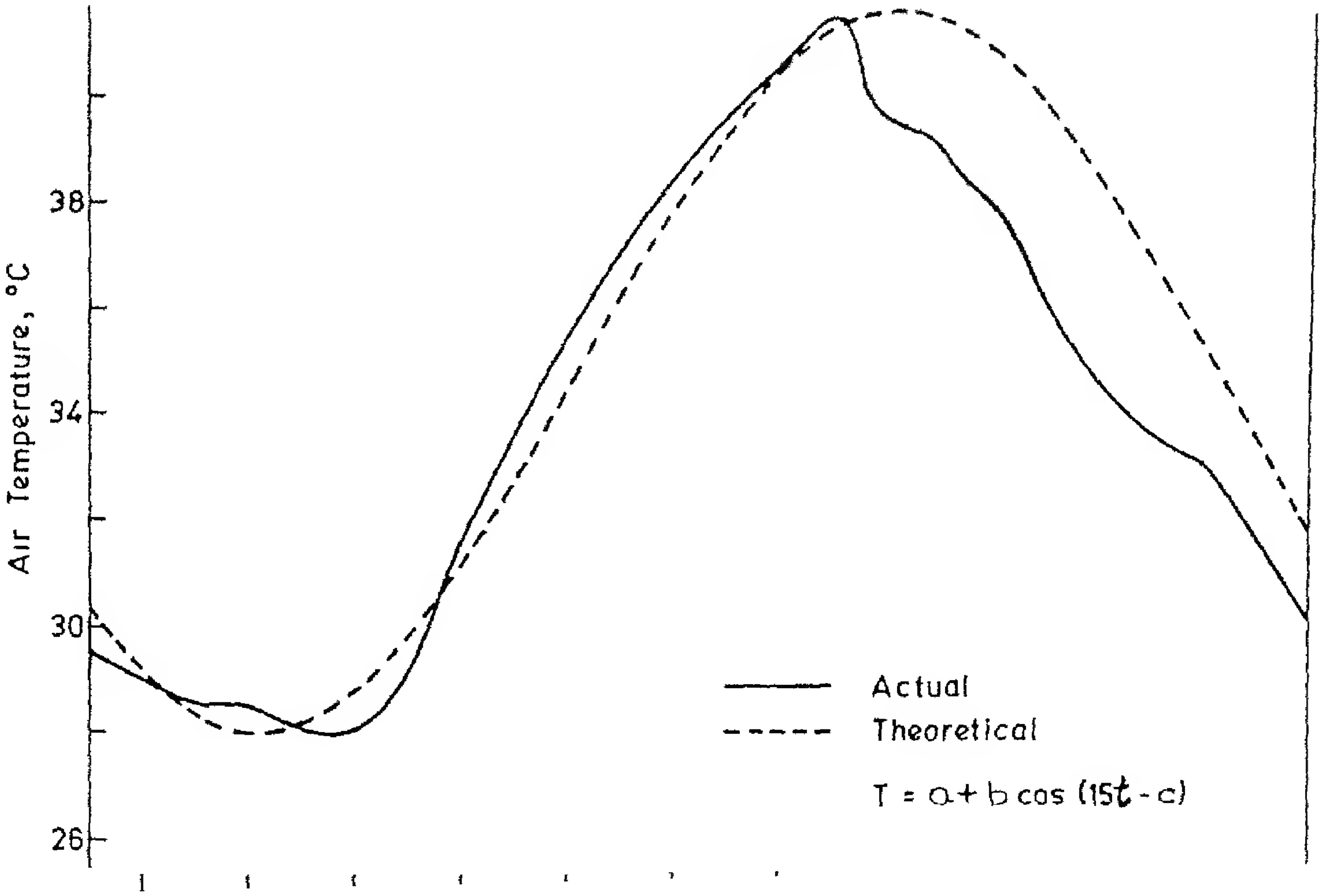
The constants a, b, c are evaluated by applying the boundary conditions as under:

$$\frac{dT}{dt} = 0 \quad \text{when } T = T_{max} \text{ or } T_{min} \tag{3.2}$$

T_{max} occurs 12 hours after T_{min}.

T_{max} and T_{min} are available for most cities in India [13].

A typical hourly variation in outdoor temperatures for Kanpur has been shown in Fig. 3.1 [8,11]. -


Air Temperature, °C
38
34
30
26
Actual
Theoretical
T = a + b cos (15t - c)

3.5 HEAT TRANSFER THROUGH STRUCTURE

3.5.1 BUILDING SURVEY

For calculation of cooling load a survey of building should be done and the following details collected:-

(a) Location of the building-longitude and latitude

(b) Orientation

(c) Dimensions of the building-length, breadth, height, thickness of each layer of the building materials used for walls, roofs and glass.

(d) Composition of building materials and their physical properties.

(e) Number of doors and windows along with shutter materials.

3.5.2 HEAT TRANSFER THROUGH WALLS AND ROOFS

The instantaneous heat transfer through walls and roof must be considered along with their thermal capacities for most building materials of finite thermal capacity, expressed as under:-

$$C_{th} = m. C_p = \rho C_p V = \rho . C_p . A.X \qquad (3.3)$$

Where, ρ = mass density (kg/m^3)

m = mass (kg)

C_p = Specific heat of building material (kJ/kg-C)

A = area of cross-section (m^2)

and X = thickness of wall or roof (m)

The thermal capacity of the wall and roof demonstrates the following two affects

(a) There is a time lag between the heat transfers from the outside surface to the inside surface (q_o and q_i respectively).

(b) There is a decrement in heat transfer due to the absorption ofheat by the wall and subsequent transfer of a part of this heat back to the outside air when ambient air temperature is lower.

Incorporating the above two effects, actual heat transfer at any time t is given by [13] .

$$\dot{Q}_t = \sum_{i=1}^{5} \Big[A_i \; U_i \; (T_{em} - T_i) . + (A_i \; U_i \; \lambda_i) . \\ (T_{o(t-\phi)} - T_{em}) \Big] \qquad (3.4)$$

Where, T_{em} = mean sol-air temperature over a day, λ is the decrement factor, ϕ is the time lag and $T_{o(t-\phi)}$ is the sol-air temperature of time $(t-\phi)$ i.e., ϕ hours before heat transfer is to be calculated. The summation symbol stands for the four walls and the roof. Figs. 3.2 and 3.3 give the values of ϕ and λ with respect to the wall thickness [6,8,16].

In case of thin walls, the decrement factor λ_i approaches unity and Eqn. (3.4) gives

$$\dot{Q}_t = \sum_{i=1}^{5} A_i U_i \left[T_{o(t-\phi)} - T_i \right] \tag{3.5}$$

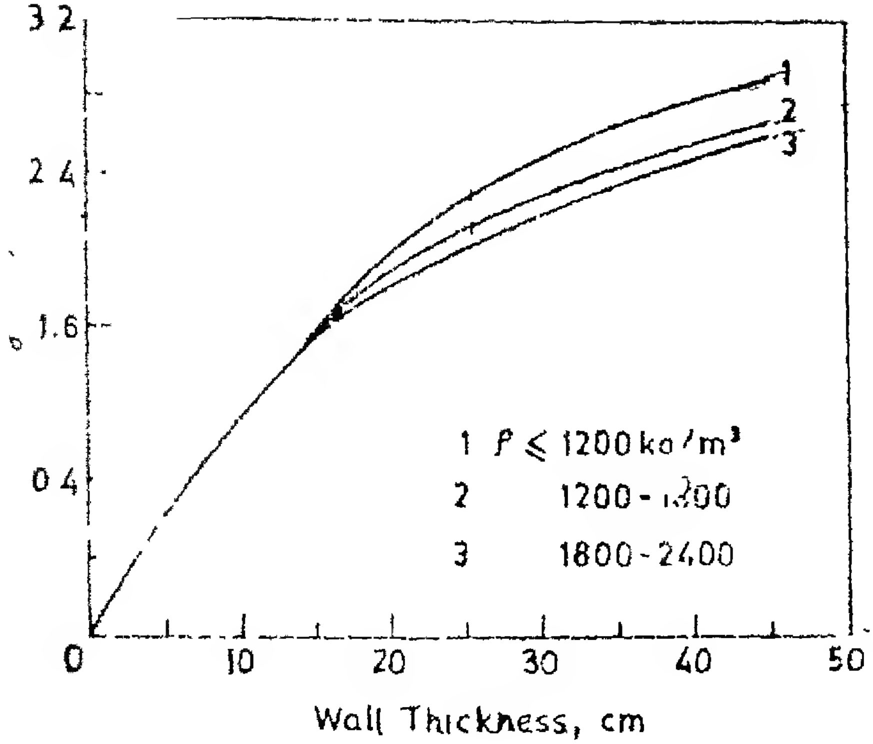
Whereas in case of very thick walls, $\lambda_i \rightarrow 0$ and hence

$$\dot{Q}_t = \sum_{i=1}^{5} A_i U_i (T_{em} - T_i) \tag{3.6}$$

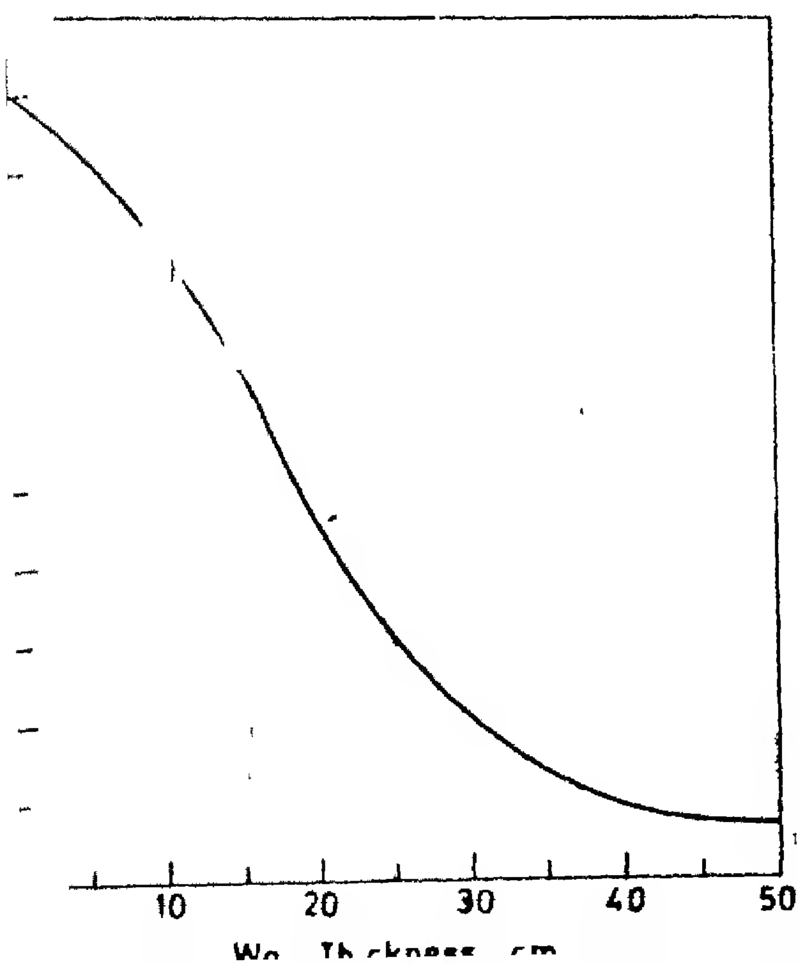
Multilayer wall and ceiling alongwith other parameters have been shown in Fig. 3.4(a) and 3.4(b) respectively. Referring to these figures, the heat conduction through structure can be written as:

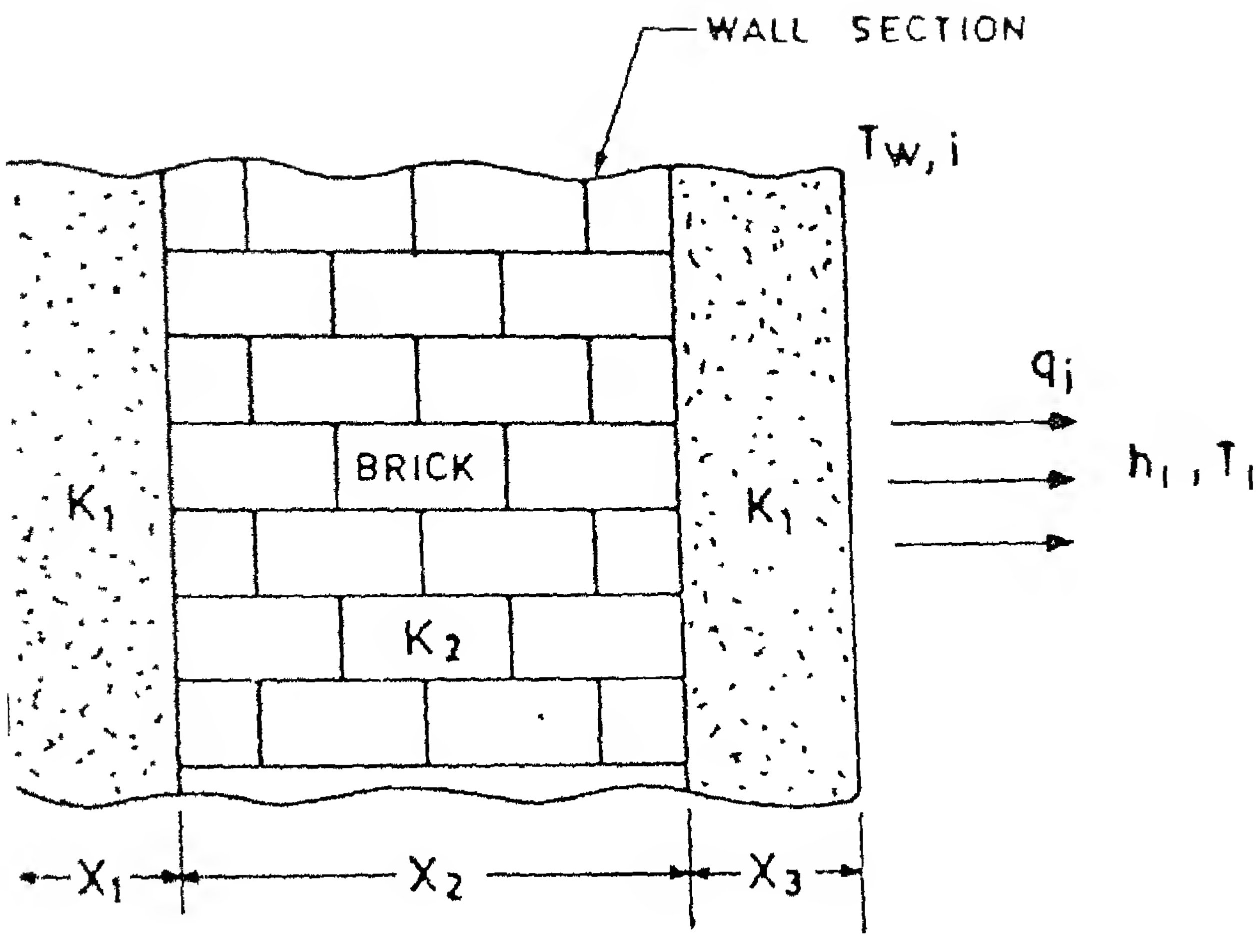
$$\dot{Q}_{st} = h_o (T_o - T_{w,o})$$

$$= \frac{(T_{w,o} - T_{w,i})}{\sum_{i=1}^{3} (X_i/K_i)} = h_i (T_{w,i} - T_i) \tag{3.7}$$



3.2 Variation in time lag with wall thickness

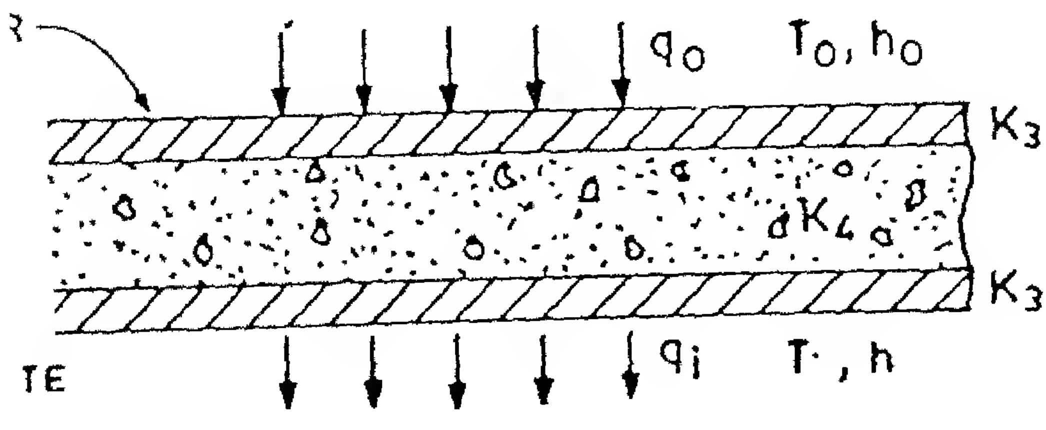




$K_1 = 0.00072\ kW/m\,°C$
$K_2 = 0.00130\ kW/m\,°C$

ig.3.4(a) Structural details of the wall.

$K_3 = 0.00072\ kW/m\,°C$
$K_4 = 0\ 00173\ kW/m\,°C$



Incorporating the overall heat transfer coefficient for walls and roof, Eqn. (3.7) can be written as:

$$\dot{Q}_{st} = U \cdot (T_o - T_i) \qquad (3.8)$$

Where, overall heat transfer coefficient U is given by

$$U = \frac{1}{\frac{1}{h_o} + \sum_{i=1}^{3} (X_i/K_i) + \frac{1}{h_i}} \qquad (3.9)$$

Where T_o and T_i ambient and inside temperatures (C)

X_i and K_i = thickness and thermal conductivity of of 'i_{th}' layer of the structural material.

h_o and h_i = convective heat transfer co-efficient for outside and inside surfaces (kW/m^2-C)

The method to get the values of h_o and h_i has been given in Appendix B.

3.5.3 HEAT GAIN THROUGH GLASS

Glass construction forms a significant part of the modern structures and it is essential to calculate the heat gain through glass. It comprises

(a) All transmitted radiations

(b) A part of the absorbed radiation that travels to the conditioned space, and

(c) Radiation transmitted due to the temperature difference between the outside and the inside environments.

Equations for the thermal properties of the glass have been given in Appendix C of 8 .

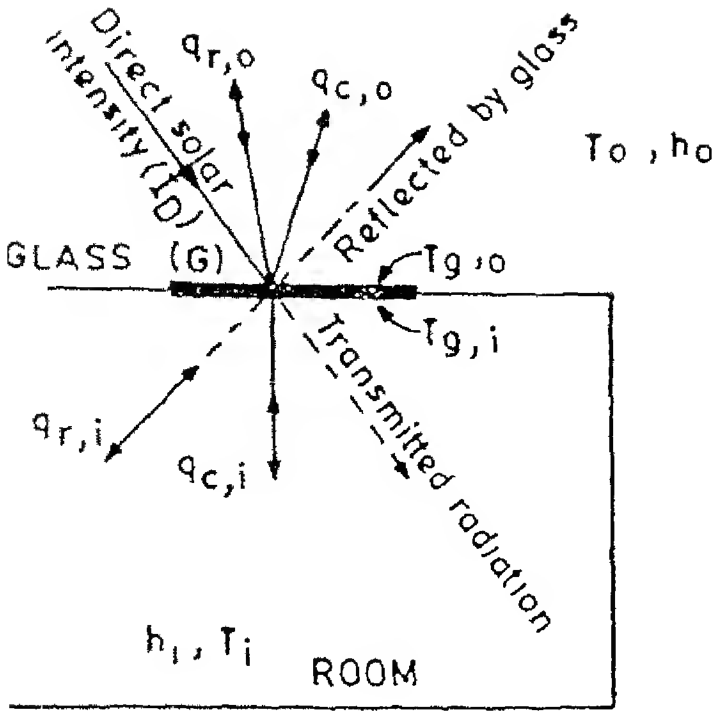
Energy balance for the glass yields:

$$\dot{Q}_{glass} = \sum_{K=1}^{N} \left[(F_s \tau_D I_D + \tau_d I_d + \tau_R I_R) + \frac{F_s a_D I_D + a_d I_d + a_R I_R}{(1 + \frac{h_o}{h_i})} + U (T_o - T_i) \right]_k \quad (3.10)$$

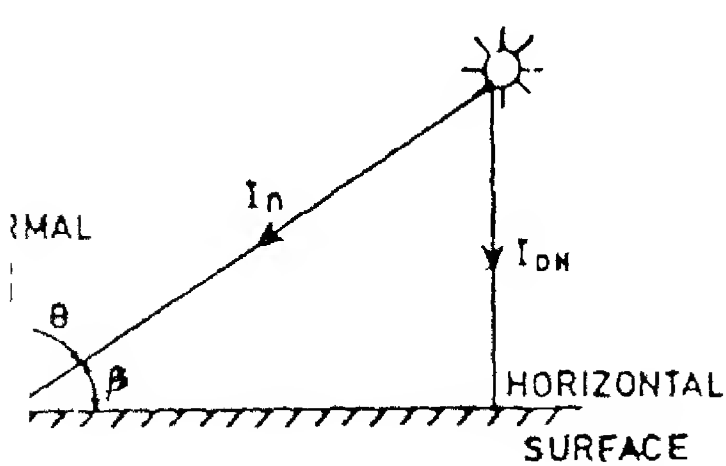
Where, τ is the transmissivity, of the glass and U is given by:

$$U = \frac{1}{\frac{1}{h_o} + \frac{X_g}{k_g} + \frac{1}{h_i}} \quad (3.11)$$

The expressions for h_i and h_o and the values of X_g and K_g are given in Appendix C of [8] whereas N refers to the number of glass windows and doors. Fig. 3.5 shows an arrangement of heat transfer through a glass window.



leat transfer through the glass window



t solar radiation on horizontal surface

3.5.4 INCIDENT SOLAR RADIATION [2, 8]

Solar radiation has very predominant effect on cooling load for the building. Its study is important for the cooling load calculation as well as to find means to reduce load and hence provide comfort conditions at economic rates. Referring to Fig. 3.6, the total solar radiation intensity (neglecting reflected radiation) is found from

$$I_t = I_{DN} \cos \theta + I_d \quad (kW/m^2) \tag{3.12}$$

Where θ is the angle of incidence of the sun's rays with respect to the normal at the surface.

I_t is the total shortwave radiation reaching the surface.

I_{DN} is the direct normal irradiation.

I_{cl} is the diffused sky radiation.

I_{DN} and I_d are given by

$$I_{DN} = \frac{A}{\exp (B/\sin \beta)} \quad kW/m^2 \tag{3.13}$$

Where, A = Apparent solar irradiation at air mass =0

B = Atmospheric extinction co-efficient

β = Altitude angle (degrees)

and $I_d = C. I_{DN} \cdot F_{ss}$ kW/m^2 (3.14)

Where C = Diffuse radiation factor

F_{ss} = Angle factor between the surface and the sky

= 0.5 for vertical surfaces

= 1.0 for horizontal surfaces

= $(1 + \cos\theta)/2$ for any other inclined surface.

The values of A, B and C are given in [8, 13].

3.5.5 SOL-AIR TEMPERATURE

The estimation of heat gain for confined space through walls and roofs involves the concept of sol-air temperature [14]. The heat balance at a sunlit surface gives the heat flux as :

$$q = \alpha \cdot I_t + h_o (T_o - T_{w,o}) - \epsilon \cdot \Delta R \qquad (3.15)$$

Where, α = Absorbtivity of the surface for solar radiation

I_t = Total solar-radiation incident on the surface (kW/m^2)

h_o = Heat transfer coefficient for long wave radiation and convection at the outer surface (kW/m^2-C)

T_o = Outdoor air temperature, C

$T_{w,o}$ = Temperature of the outside surface, C

ϵ = Hemispherical emittance of the surface

= 1 for blackbody

ΔR = Difference between the longwave radiation incident on the surface from the sky and surroundings and the radiation emitted by a blackbody at outdoor air temperature, kW/m^2

= 0.063 kW/m^2 for horizontal surface [2, 8]

= 0 for vertical surface [2,8]

The value of I_t in Eq. (3.15) is determined by the fundamental angles, i.e., the sun's declination angle (d), latitude angle (l) and the hour angle (d_n). Certain other angles used in solar radiation calculations which can be desired in terms of the fundamental angles are the sun's zenith angle (Ψ), altitude angle (β) and azimuth angle (γ).

In terms of sol-air temperature (T_{sol}) Eq. (3.15) can be written as:

$$q_o = h_o (T_{sol} - T_{w,o}) \qquad (3.16)$$

Where, $$T_{sol} = T_o + \alpha . I_t / h_o - \epsilon . \Delta R / h_o \qquad (3.17)$$

The outside air convection heat transfer coefficient h_o [8,17] is determined as under:-

$$h_o = 0.00391\ (0.399 + V_w)\ (kW/m^2\text{-}C) \text{ for eterior walls}$$

$$= 0.00141\ (1.454 + V_w)\ (kW/m^2\text{-}C) \text{ for roofs} \quad (3.18)$$

Where V_w = Velocity of outside air (m/s)

3.6 VENTILATION LOAD

In order to maintain required levels of oxygen, carbon-di-oxide as well as odour level and to set up satisfactory conditions for occupants certain quantity of outside air is introduced into the confined space. The ventilation requirement varies according to usage (for operation rooms in hospitals 100% air is ventilated). The ventilation codes and standards recommend a minimum rate of 30 cfm /person. But recent studies in the light of energy conservation show that fresh air at a rate of 5 cfm/person is sufficient because of economic consideration [18] . Ventilation load is given by :

$$\dot{Q}_{vent} = \frac{\text{No. of occupants}}{v_a} \times \text{ventilation rate per person } (h_{oa} - h_{ia}) \quad (3.19)$$

Where, v_a = Specific volume of air, m^3/kg of dry air.

h_{oa}, h_{ia} , = enthalpies of outside and inside air, kJ/kg of dry air.

3.7 INFILTRATION LOAD

Infiltration of the outside air into the confined space occurs in almost every building through cracks and openings being caused by pressure difference between the outside and inside air. The natural infiltration will supply enough air to meet the ventilation requirements for a residence or a small office where there are no objectionable odours inside the space. However, in actual practice whatever be the amount of infiltration air, the ventilation air is definitely supplied as per usual practice. Infiltration of air may cause sensible as well as latent heat load for the confined space and is given by

$$\dot{Q}_{infit} = \frac{V_{room}}{v_a} \times N_{ACH}\ (h_o\ a - h_i\ a)/(24 \times 3600),\ W \quad (3.20)$$

Where, V_{room} = Volume of the conditioned space, m^3

N_{ACH} = Number of air changes required per day.

The volume of infiltration is related in terms of the volume of the room and tabular values are available in [1,2] .

3.8 LIGHTS AND OTHER ELECTRICAL APPLIANCES

The load due to electrical appliances comprises various electrical items like lights, fans, freezer, motors etc.

$$\text{Hence } \dot{Q}_{elect} = (\text{No. of appliances x rating}) + \text{No. of lights x rating}) \times \text{Cooling load factor} \quad (3.21)$$

Cooling Load Factor (CLF) is a function of time of use, type of arrangement, room furnishings etc. It is quite safe to select its value to be 0.88 for light load. However, the actual power input to the light point should be taken.

3.9 HEAT GENERATION INSIDE THE CONDITIONED SPACE

This consists of heat generation from human beings inside the space and product load. The load from human beings

is calculated based on the numbers and the various activity levels. It is given as:

$$\dot{Q}_{occu} = \sum_{j=1}^{n'} N_{Pj} \times h_{Gj} \tag{3.22}$$

Where N_{Pj}, h_{Gj} and n' are number of persons and corresponding heat generation per person of the jth type of activity.

The total cooling load is calculated from :

$$\dot{Q}_{total} = \dot{Q}_{str} + \dot{Q}_{infil} + \dot{Q}_{vent} + \dot{Q}_{elect} + \dot{Q}_{occu} \tag{3.23}$$

CHAPTER - 4

SYSTEM DESIGN AND EXPERIMENTAL SET-UP

4.1 INTRODUCTION

The design in itself is a challenging subject, dealing with a purposeful activity directed towards the fulfilment of human needs. This requires considerable informations such as feasibility of certain systems within the limiting factors namely physical realizability, economic worthwhileness, financial feasibility etc. In another way, it can be said to be a complete interpreneourship. In addition one has to consider factors such as materials, component and their design steps, components and their optimum combination, besides its adoptability by customers.

In tropical countries the outside and inside design conditions do support the use of evaporative coolers from April to June and conventional airconditioners from July to October . Hence for such type of cooling loads, the Hybrid System of airconditioner offers a good promise. This has

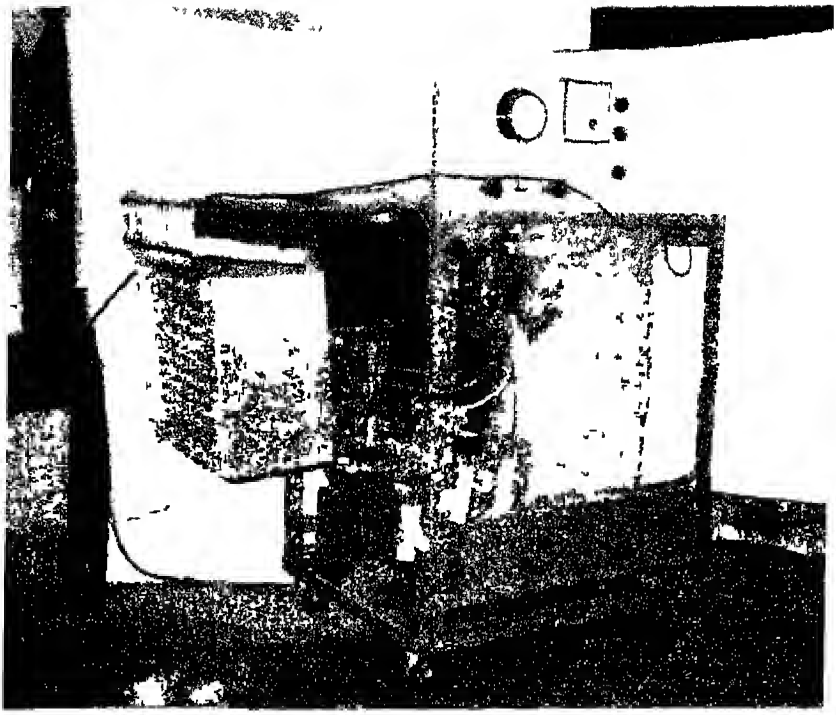


Fig. 4.1 A hybrid airconditioning system.

motivated us to develop a prototype of a 1.5 ton Hybrid Airconditioner using R-22 as the refrigerant. The system integrates parts of conventional airconditioner and evaporative cooler as under :-

(a) Parts of conventional airconditioner used

(i) Compressor

(ii) Evaporator

(iii) Expansion device

(b) Other parts used

(i) Evaporative Condenser

(ii) Blowers

(iii) Water circulation sub-system

(iv) Instruments and controls

4.2 DESIGN/SELECTION OF SYSTEM COMPONENTS [19]

Figure 4.1 shows the photograph of the prototype of 1.5 ton Hybrid Airconditioning system. All the components have been mounted on 55x40 mm slotted iron frame work and supported on caster wheels for easy movement from one place to the other. To achieve standardization and easy availability of the components in the market, mostly standard

equiomment have been used. The considerations for selection/ design of components are given in succeeding paragraphs.

4.2.1 SELECTION OF REFRIGERANT

Refrigerant -22 ($CH\ Cl\ F_2$) has been selected because of lower boiling point (232 K i.e, - 41C) and enthalpy of vapourization at about 1 bar being 233.69 kJ/kg (about 134% higher than R-12). It is a non-toxic non-inflammable, non -corrosive and non -irritating refrigerant.

4.2.2 SELECTION OF COMPRESSOR

The purpose of compressor is to maintain a desired evaporator pressure corresponding to the requirement of low temperature. It should take less power within a wide range of operation. It should give trouble free service for a long time, be cheap and require less maintenance. In the present development S1622 compressor of Shriram Refrigeration Ind. Ltd. with the following specifications has been used

Input power	2200 W
Displacement volume per revolution	40.8 cu cm

RPM	2850
Evaporator temperature	7.2 C
Condenser temperature	55 C
Evaporater temperature range	− 3.9 to 12 C
Ambient temperature	35 C

4.2.3 EVAPORATOR SELECTION

An evaporator should transfer enough heat from as small size as possible. The liquid should not leave the evaporator in order to prevent the wet compression. The liquid supply to evaporator should be easy. The size and arrangement of the pipe should be so adjusted as to cause easy oil return to the compressor crank case. The corrosion and fouling of the inside and outside surfaces should be minimum. It should be light, compact, safe and durable. The pressure loss should be as low as possible.

In the prototype of Hybrid airconditioner a 1.5 ton cooling capacity Fin and Tube type evaporator has been used. The specifications of the evaporator are as under :

Number of tubes	=	20
Tube outside diameter	=	1.25 cm

Tube length	=	53 cm
Fin pitch	=	4.4 per cm
Fin thickness	=	0.04 cm
Frontal area	=	53 x 22.5 cm
Fin length	=	22.5 cm

4.2.4 SELECTION OF EXPANSION DEVICE

The expansion device is one of the basic components of a mechanical refrigeration system. It is installed between receiver/condenser and evaporator. The expansion device reduces the condensate pressure down to evaporator pressure and regulates the flow rate through evaporator. These can be broadly classified as valve type and capillary type.

In the present case a capillary tube with the following specifications has been used.

Length of capillary tube	=	174 cm
Tube outside dia	=	0.256 cm

4.2.5 SELECTION OF BLOWERS

Blower is a device to produce flow of air. These are identified into two general groups centrifugal and axial flow blowers. In the present prototype two blowers

Acc No. A 105916

were required one for sucking the ambient air through the wetted pad and the other to feed the conditioned air (either through wetted pad or through evaporator) to the confined space. Based on the requirement following type of blowers were selected.

(a) Gulmarg Blower:-

This centrifugal type radial blade blower because of its good suction from both ends was selected to suck the air through wetted pad. Its specifications are as under :-

Make	Gulmarg
Speed	1400 RPM 940 RPM
Discharge Area	30 x 27 cm
Motor	125 - 150W

(b) Evaporator Blower :-

It is centrifugal type radial blade blower which has been selected for the following purpose :

(i) In the evaporative cooling mode of operation, it increases the velocity of the air being fed by the Gulmarg blower.

(ii) In the airconditioning mode of operation it draws the air through the evaporator and feeds it to the room. Its specifications are as under :

Speed	1970 RPM
Voltage	230 AC
Motor	150 W

4.3 DESIGN OF EVAPORATIVE CONDENSER UNIT

Condenser is a heat exchanger in which the desuperheating of high temperature vapour, phase change (from vapour to liquid) and subcooling of condensate occur. In the present system a suction or induceed type evaporative condenser has been used. Condenser coils have been embedded in wood wool pad. Water issued from the top trickles down over the cooling coil and is cooled by both evaporatively cooled air and spray of water. The condenser unit is capable of working in airconditioning as well as evaporative cooling mode.

4.3.1 COIL LENGTH AND WET PAD

The thermodynamic vapour-compression cycle simulating the refrigeration of the airconditioner is shown in Fig. C.2.

According to operating conditions given in the pamphlet of Kirloskar Brothers Ltd., for 1.5 ton system operating on R-22 having T_h = 54.4C the heat rejection is 6.37 kW. Since in the present system the condensing coils are evaporatively cooled, the condensing temperature is assumed to be 46C. By carrying out thermodynamic analysis based on actual vapour compression cycle (as given in Appendix C) the heat rejection has been found to be 6.548 kW which is in agreement with the analysis carried out by Mishra [8]. Analysis carried out on the basis of evaporator and condense pressures found experimentally reveal heat rejection of 6.548 kW. The analysis based on inlet and outlet conditions of air to the condenser unit in airconditioning mode gives heat rejection as 3.55 kW. These results are shown in Appendix D .

Extensive Experiments carried out by Mishra [8] revea the heat rejection per m^2 of the wetted pad condenser to be 6.42 kW/m^2. He had computed the length of 9.525 mm copper tube for the condenser for 1.5 ton hybrid system to be 67.62 m using wood-wool density of 0.157 gm/cm^2). In the present case a better porous pad (wood-wool 0.1 gm/cm^2) of 1.5x1 m^2 has been used. The blower used for the condenser unit is of a larger flow rate, thus giving better convective heat transfer. Accordingly a copper tube of 9.525 mm diameter and 42 m length has been used.

The conventional air cooled condensers have much less face areas as compared to wetted pad condensers because the latter have no fins. Similarily the copper tube length of the hybrid system has gone up. This increase in the cost due to larger copper tube length is compensated by eliminating aluminium fins [20] .

The air-flow rate in the evaporative cooling mode is found to be 370x0.51x0.1 = 37.74 m^3/min. For Kanpur the summer design condition is T_{db} = 41.2 C and T_{wb} = 21.5 C. This renders the temperature of air leaving the evaporative cooler at

$$T = 41.2 - 0.75\ (41.2 - 21.5) = 26.43\ C.$$

Using 5 C as temperature rise, this will meet a cooling load of (37.74/ 0.88) x 1.026 x 5/60 = 3.67 kW. As there is no infilteration and ventilation load and the temperature difference across wall is not much due to the maintenance of higher inside temperature, one may feel comfortable.

4.3.2 SELECTION OF PUMP

A Tullu Vijay monobloc pump has been used to spray water on the pad in order to cool air for the evaporative cooling mode of operation and for dissipating heat of compressed vapour by evaporatively cooled air and water.

The pump supplies water at a rate of 0.16 kg/s for spray over the pad . The pump specifications are :

LMP	400
Head	1.25 m
Current	0.35 Amp
Voltage	230 V AC

4.4 FABRICATION OF THE SYSTEM

4.4.1 FRAME WORK

The system comprises a compressor, an evaporator, a blower for evaporator, a wood-wool pad containing copper tubes of the condenser, a blower for the condenser, a pump,

a water tank, electric control board. In order to render flexibility in this development, the slotted angle iron structure has been used. This structure has dimensions of 950x800x1150 mm using 55x40 mm slotted angles. The frame has been supported on four 100 mm high caster wheels. A 3 mm GI water tank of 800x700x200 mm size has been supported on the bottom of the frame work with the help of 55x40 mm slotted angle cross numoers. Supports for compressor and evaporator have been provided appropriately to render compact structure.

4.4.2 COMPRESSOR

A SR 1622 Sriram Industries compressor has been used for the system. The specification of the compressor have been given in sec 4.2.2.

The inlet and outlet connections of the compressor has been brazed (using copper eutectic) to evaporator and condenser coil respectively.

4.4.3 EVAPORATOR

A conventional 1.5 ton evaporator has been used. The specifications have been given in Sec. 4.2.3.

4.4.4. EXPANSION DEVICE

A copper capillary tube having 174 cm length and 0.256 cm outer diameter has been used. The capillary tube has been connected to the condenser coil using reducer and 5mm copper tube.

4.4.5 CONDENSER

The condenser unit for the hybrid airconditioner has been fabricated in the Refrigeration and Airconditioning Laboratory at Indian Institute of Technology, Kanpur. The details of the fabrication are given below :

(a) A 9.525 mm copper tube having length of 42 m as determined in Sec 4.3 was bent into 'U' shape coils. These coils were then mounted on a frame work of wiremesh and secured with the help of GI wire. Two wood-wool pads at the rate of 0.05 gm/cm^2 were made and secured on both sides of the coils so as to form uniform wetted surfaces. One end of the coil was brazed to the compressor outlet and the other to the capillary tube inlet.

(b) This condenser unit was suspended in a tank in such a way that about two lengths of each coil would remain in water to provide subcooling of the condensate. The arrangement of condenser unit is shown in Fig. 4.2. An aluminium cover 1 mm thick is put over the condenser unit to prevent suction of fresh air from atmosphere.

4.4.6 WATER CIRCULATION SYSTEM

The water circulation system for the hybrid aircondi-tioner consists of the following:

(a) GI Tank

A GI tank of 800x700x200mm size is used for storing and collecting the spray of water. It is supported on the slotted angle frame in such a way that the tank can be removed for cleaning without interfering with other components.

(b) Pump

A Tullu Vijay pump has been used for feeding water from the tank to the spraypipe. The pump is capable of delivering 200 LPM at a head of 1.25 m.

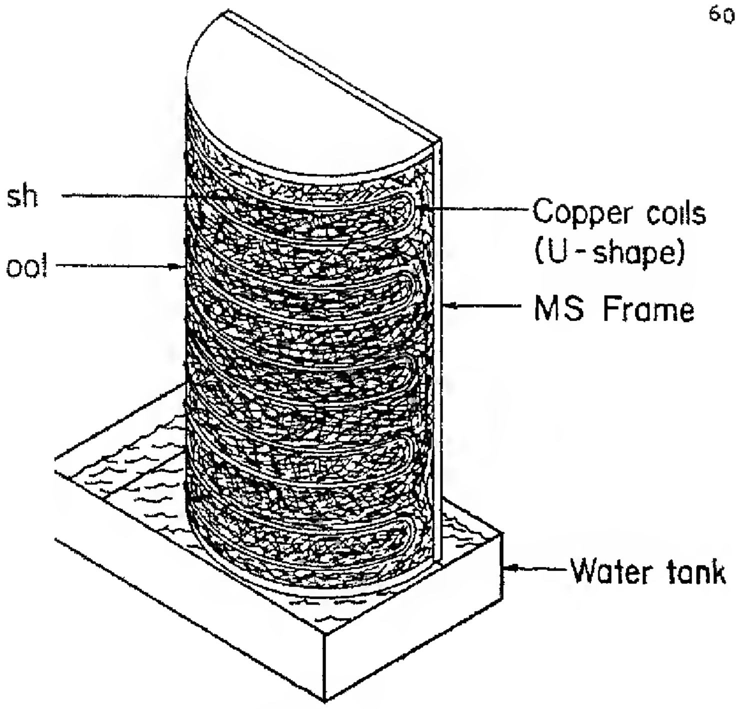


Fig 4.2 Condenser unit

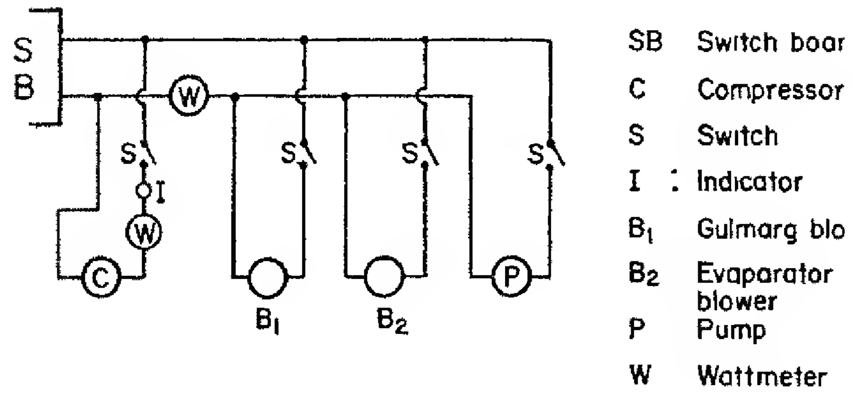


Fig 4.3 Electric circiuit diagram

(c) Spray Pipes

Two copper pipes of 15.6 mm diameter and 1.5 m length have been mounted above the condenser coils. 2mm diameter holes spaced at 4 cm have been drilled to spray water on the pad. One end of the pipes is connected to the pump and the other closed with wooden plugs.

4.4.7 BLOWERS AND DUCTING

Two blowers have been fitted in the system. A two speed Gulmarg (centrifugal radial vane type) blower has been placed inside the 'U' shaped condenser cage. It has been supported on an angle iron frame one side and a wooden prop (resting in the tank) on the other. This blower sucks air through the wetted pad and feeds it to the room/discharges it to the atmosphere based on the mode of operation.

The other blower operates in airconditioning mode. It draws air over the evaporator coil and supplies to the room after cooling and dehumidification.

Suitable ducting of 1 mm aluminium sheet as shown in Fig.4.4a has been used. A flap incorporated in the duct allows the air sucked by the Gulmarg blower to either be fed to the room (in evaporative cooling mode) or discharged to the atmosphere (in airconditioner mode).

4.4.8 INSTRUMENTS USED

Instruments used in the system are as under :

(a)	Pressure gauge	(0 - 300 PSI)
(b)	Wattmeter	(0 - 5000 W)
(c)	Thermometers	(0 - 120 F, 0-50C , 0-100C)
(d)	Psychrometer	(0 - 100 C)
(e)	Anemometer	(Vane type, 0 -1,00,000 ft)
(f)	Manometer	(digital, 0 - 20mm, least count .001 mm) of water

Controls used in the system are as under :

(a) Valves for low and high pressure connection to the pressure gauge.

(b) Valves for charging the system with refrigerant.

(c) Switches for operating compressor, blowers and pump.

(d) Indicator lights for compressor operation, evaporative cooling mode and airconditioner mode of operation.

4.4.9 ELECTRICAL CIRCUIT

A 230 V single phase stabilized electric supply was given to the switch board of the system. The power was distributed to compressor, blowers and pump. A diagram showing the electrical system is given in Fig. 4.3.

4.4.10 MISCELLANEOUS

The duct and pipe leading from evaporator to compressor have been wrapped with foam insulation sheet of 6 mm thickness. In order to reduce the heat gain to the cool air 2.54 cm thick thermocole was also wrapped round the supply and distribution ducts. The M.S. sheet grilles are provided in the suction line and a filter pad to prevent the dust and suspended impurities going to the evaporator coil. The M.S. sheet discharge grilles are adjusted to give uniform air distribution as well as spread of air as per desire. The velocity of air at grille was kept at 3-5.5 m/s. The suction velocity is kept as 0.6 m/s. This almost eliminates the hissing sound due to suction of air.

4.5 SYSTEM CHARGING

First the system was checked for leakage by supplying air pressure up to 12 atm. The leakage was detected using soap solution and all leaks were removed by rebrazing and

tightening of joints. Thereafter the system was left under pressure for more than 24 hours to ensure completely leak-proof system. Then the system was evacuated to almost zero vacuum and it was tested for leakage under vacuum for 24 hour duration. After this the system was purged with R-22 refrigerant and evacuated twice such that the system is free from air and other non-condensable gases including water vapour.

Finally the system was charged with R-22 such that the rated current is taken by the refrigeration system. The head pressure was also checked during this process. Finally the system became ready for operation.

4.6 SYSTEM OPERATION

4.6.1 EVAPORATIVE COOLING MODE

To provide thermal relief during hot dry climates the system can be operated in evaporative cooling mode. Water is sprayed on the wetted pad with the help of pump and spray pipes. When the wood-wool pad gets thoroughly

wet, the blower placed inside the condenser unit is started. It sucks the air through the wetted pad where it gets evaporatively cooled. Thus the temperature is reduced and some moisture is added for comfort conditioning. This air is then fed to the room for once through cooling. The power consumption in this mode is around 300W. A line diagram depicting this mode has been given in Fig. 4.4(a). Higher air circulation can be used in the room if the ambient conditions are too severe.

4.6.2 AIRCONDITIONER MODE

In case of hot-humid climates, thermal comfort is provided by operating the system in airconditioning mode. Water is sprayed on the condenser unit and once this gets sufficiently wet, blower is started to provide evaporative cooling to the condenser coils. The exit air is discharged to the atmosphere. Compressor is put on, which compresses the refrigerant vapour and feeds it to the condenser. In th condenser it is cooled followed by sub-cooling (due to immersion of the lower coils in water). The sub-cooled refrigerant is passed through the capillary tube which reduc the pressure of the refrigerant. Then this refrigerant is s to the evaporator where it vaporizes by absorbing heat from

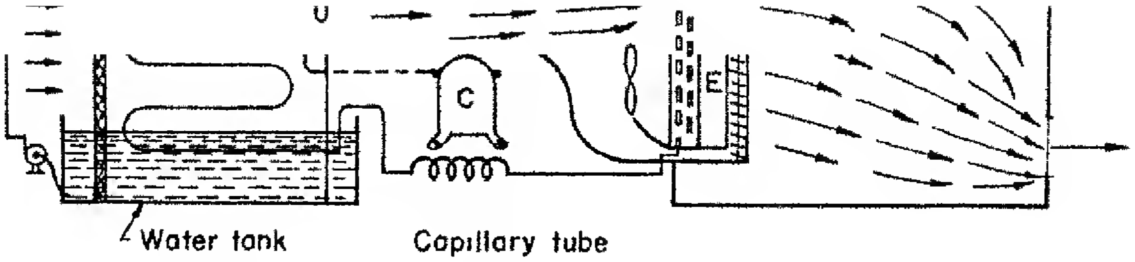
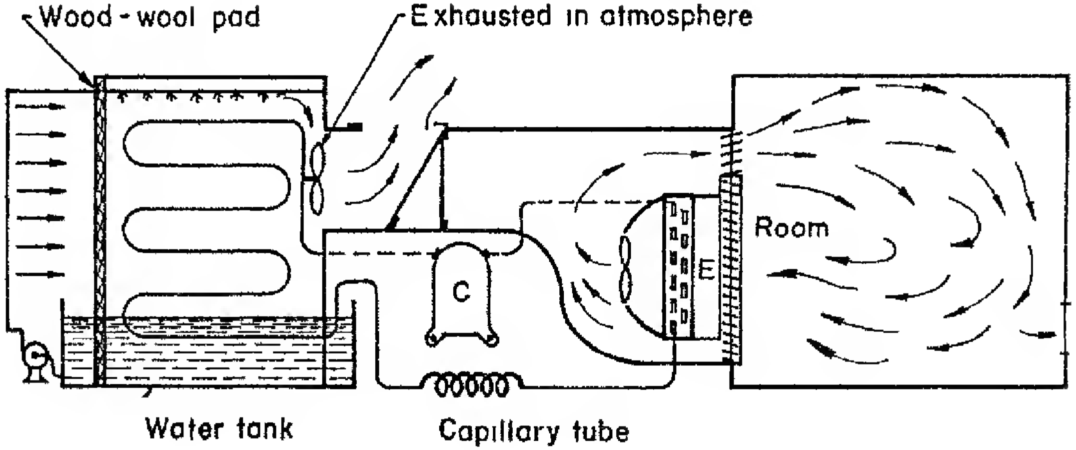


Fig 4.4 (a) Hybrid system in evaporative cooling mode



air being passed over it, rendering cooling effect. In the upper coils the vapour gets superheated and is fed to the compressor (avoiding wet compression). Total power consumption in this mode is around 1600 - 1700 W. The layout of Airconditioning mode operation has been shown in Fig. 4.4 (b).

CHAPTER - 5

RESULTS AND DISCUSSION

The developement of the Hybrid Airconditioner has been done with a view to consume lesser energy for comfort airconditioning and thereby saving power for higher priority sectors like industry and agriculture. In order to design the hybrid system the condensing unit has been modified to serve dual objectives :

(a) to work as an evaporative cooling device for thermal relief during hot-dry climates and

(b) to work as evaporatively-cooled condenser during airconditioning mode of the system operation.

Thermodynamic model has been simulated based on actual vapour-compression cycle and the effects of condensing temperature and evaporator temperature on refrigeration effect, heat rejection by condenser, input power to compressor and COP have been analysed.

The system has been operated in airconditioning mode. Its performance has been obtained based on the variation in ambient conditions, evaporator and condenser pressures.

Various quanties have been calculated and discussed in detail.

5.1 THERMODYNAMIC ANALYSIS

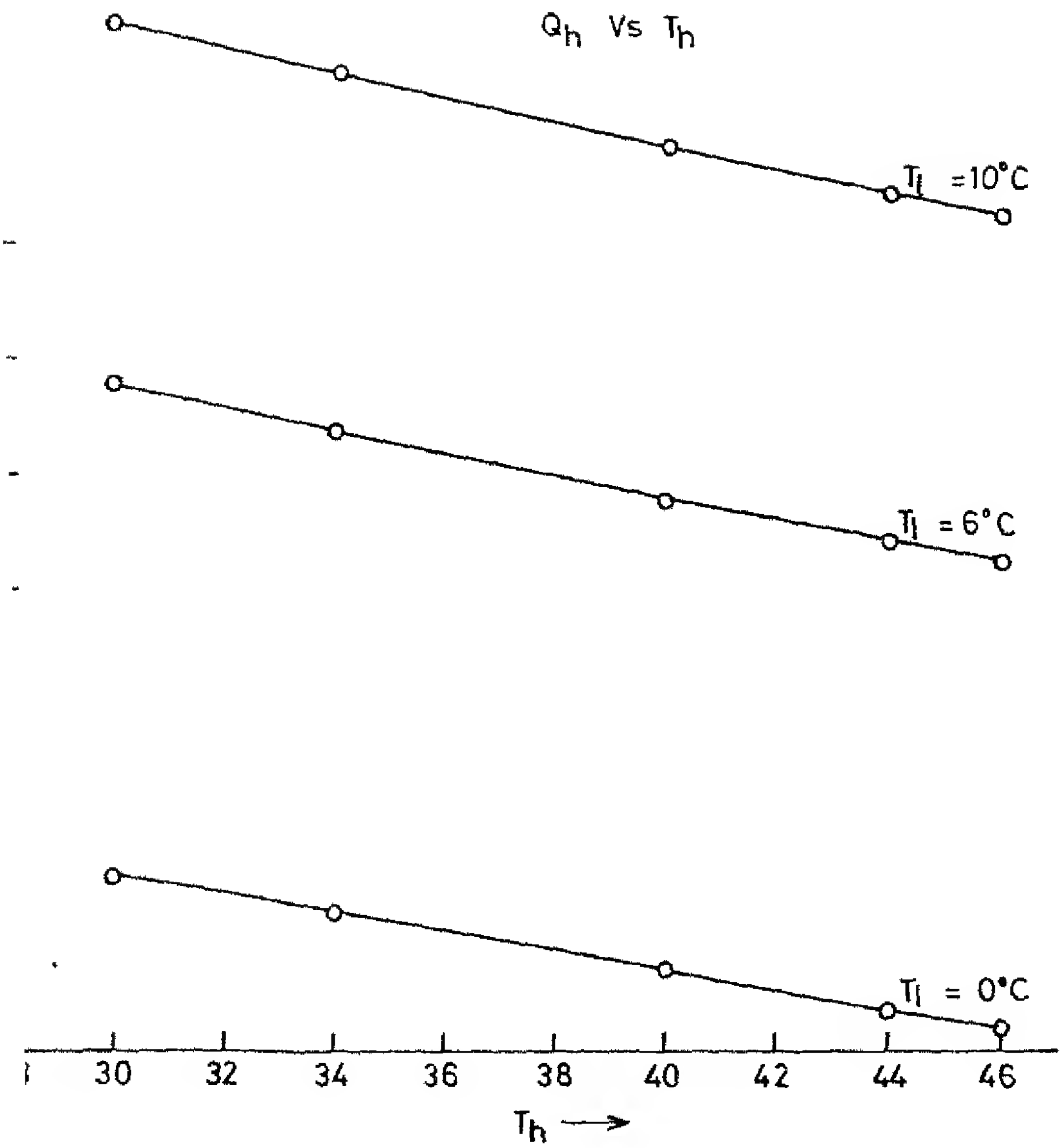
To get an idea of performance of the actual vapour compression system, the thermodynamic model has been simulated to include pressure drops during suction and discharge of compressor, vapour superheat in the evaporator and subcooling of condensate in order to see the performance with variable parameters of condensing and evaporating temperatures.

5.1.1 VARIATION OF HEAT REJECTION

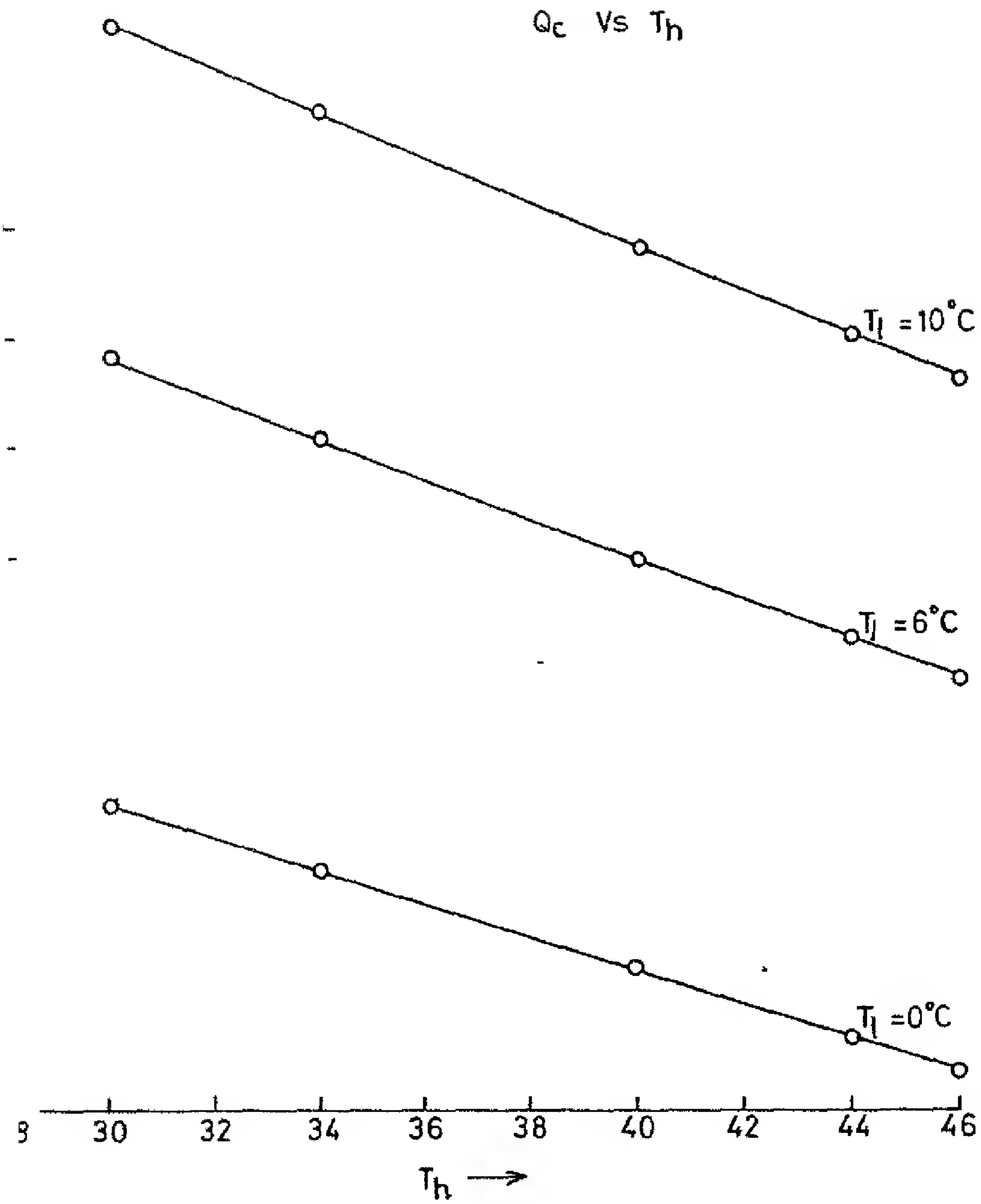
Figure 5.1 shows the heat rejection from condenser with respect to condenser and evaporator temperatures. Heat rejection reduces slowly with rising condensing temperature because of the reduction in temperature gradient available for heat transfer. It varies directly with increase in evaporator temperature.

5.1.2 VARIATION OF REFRIGERATION EFFECT

This variation has been depicted in Fig. 5.2. Refrigeration effect reduces with rise in condensing temperature and increases with rise in evaporator temperature.



Variation of heat rejection with condensing temperature



/ariation of refrigeration effect with condensing temperature

5.1.3 VARIATION OF COMPRESSOR POWER

Compressor power increases with increasing condenser temperature since the compressor has to compress the refrigerant to higher pressures. This power also increases with increase in evaporator temperature as the compressor has to handle greater volumes of refrigerant vapour due to increase in specific volume (because of temperature rise). This variation has been shown in Fig. 5.3.

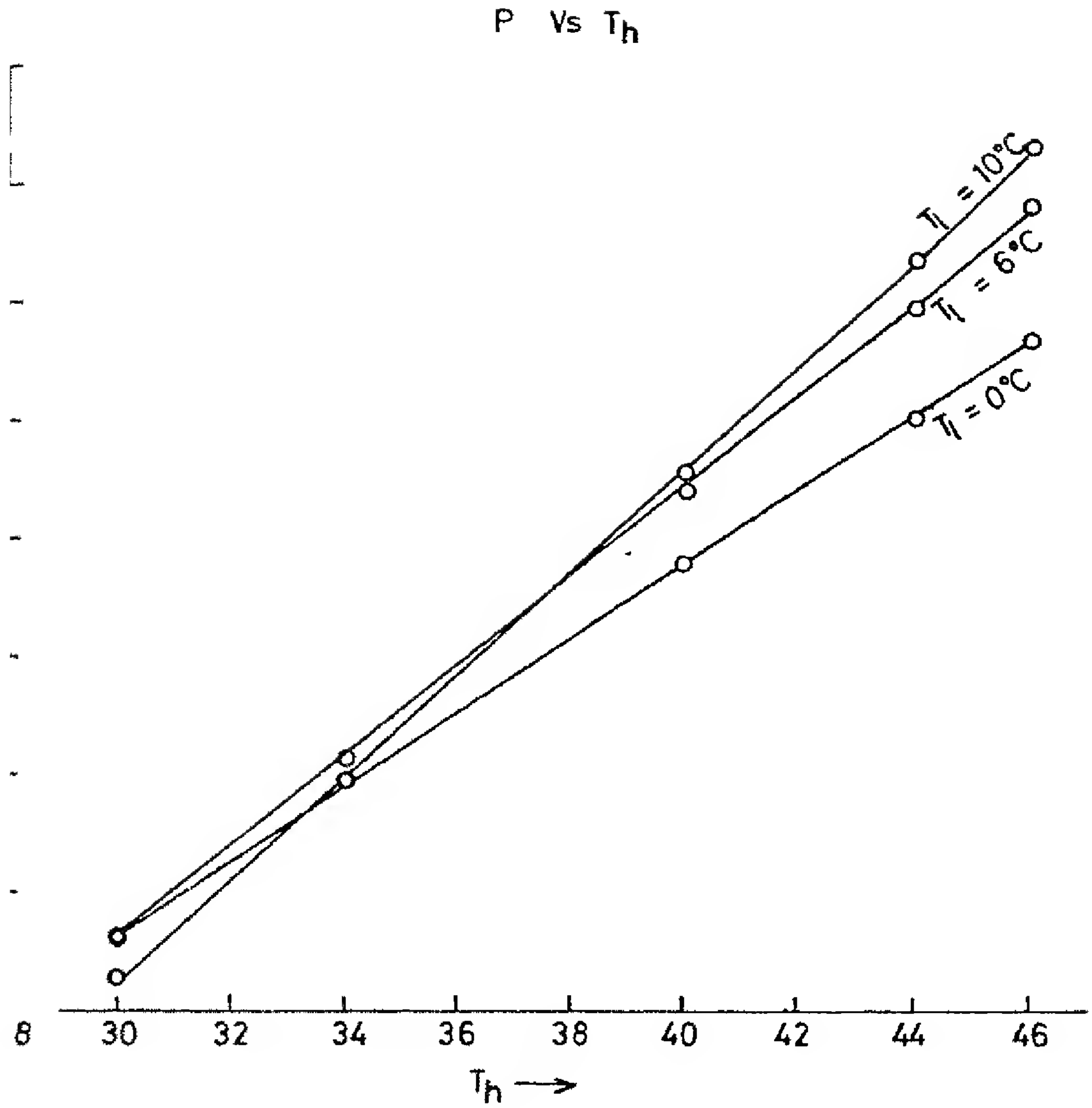
5.1.4 VARIATION OF COP

COP is defined as the ratio of cooling effect produced to the work input to the system. In the theoretical analysis, COP is found to be reducing with increasing condensing temperature (since $\dot{Q}_c$ reduces and power input to compressor increases). However, it is found to rise with the rise in evaporator temperature. COP variation has been depicted in Fig. 5.4.

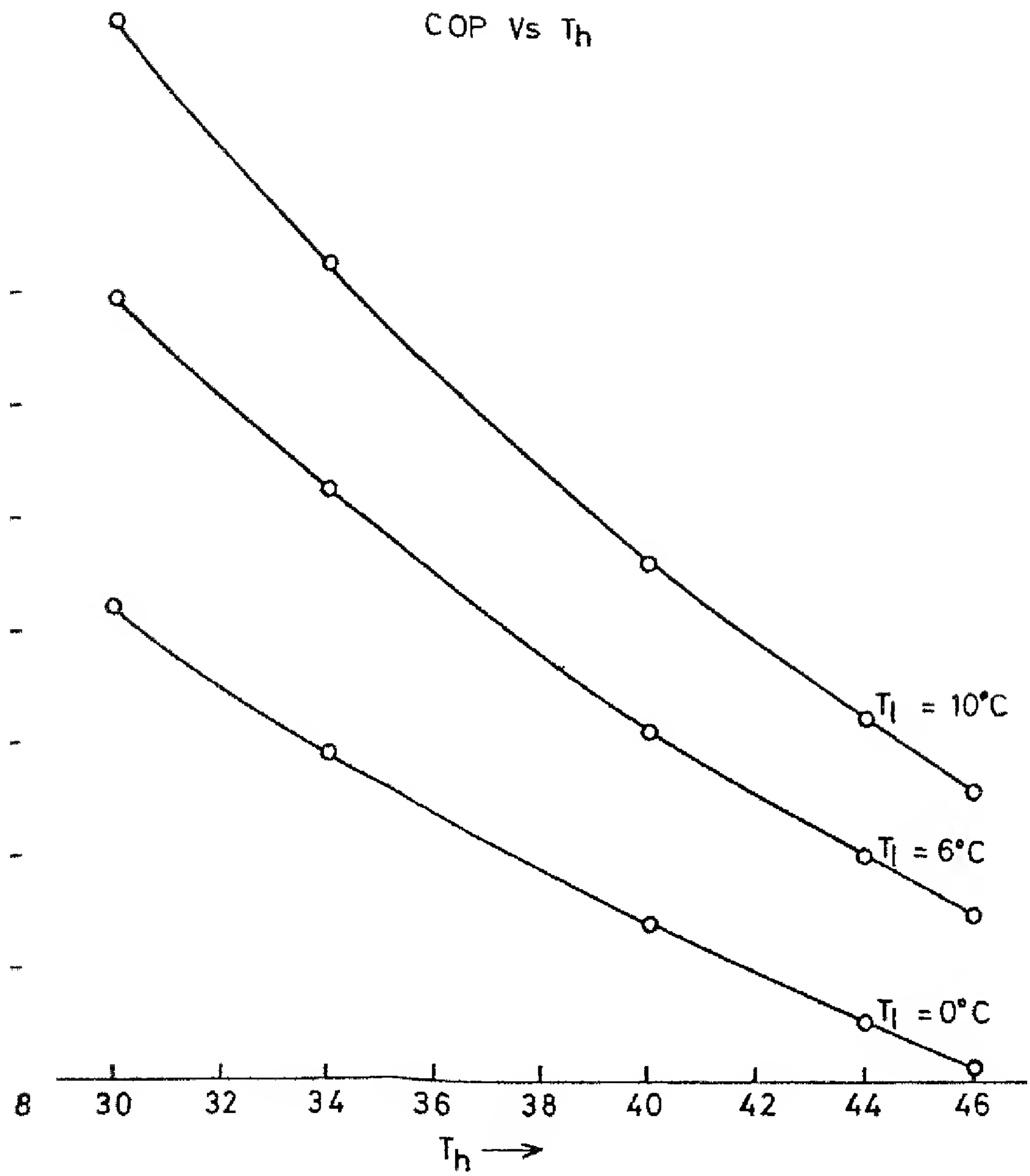
5.2 SYSTEM PERFORMANCE

The hybrid airconditioner system was operated in the airconditioning mode to provide thermal comfort in hot and humid climates. The variations in various parameters have been as under:

(a) Ambient Air conditions varied from T_{db} = 29 C to 33 C and relative humidity from 78% to 92%.



5.3 Variation of power with condensing temperature



Variation of theoretical COP with condensing temperature

However, the rainy season design conditions for Kanpur is reported to be T_{db} = 32.1 C and T_{wb} = 28.1 C in [17].

(b) Condenser and evaporator pressures vary in the range of 18.10 to 19.48 bar and 4.80 to 5.97 bar respectively and render cool operation of the compressor giving it longer life and trouble free operation.

(c) Condition of the air being discharged from the Gulmarg blower (placed in condenser unit) varied from T_{db} = 33 C, ϕ = 96% to T_{db} = 29 C, ϕ = 89%.

(d) Compressor Power for the SR 1622 model varied from 1.275 kW to 1.400 kW for different loads. The power requirement for blower and pump was 290 W. Thus the total power variation turns out to be 1565 to 1690 W.

(e) Temperature variations in the discharge air from grille were found to be from 17 C to 20.5 C and relative humidity from 86% to 98%.

(f) By using the hybrid airconditioner, the inside conditions obtained in the room varied from

T_{db} = 23 C, ϕ = 80% to T_{db} = 27.0 C and ϕ = 78%. These conditions were found to be quite comfortable by the occupants.

The above results have been given in Table 5.1.

5.3 ESTIMATION OF REFRIGERATION EFFECT AND COP

Based on the experimental readings the refrigeration effect and COP were calculated. The refrigeration effect varied from 3.888 kW to 5.15 kW (i.e; tonnage of the system 1.115 - 1.4717) and COP from 2.385 to 3.29. These results have been tabulated in Table 5.2 . The actual values are found to be lower than that of the rated capacity of the system. This is due to the fact that the evaporator temperature was not allowed to go to much lower value as is normally used. Secondly, the system gave the desired inside conditions after consuming much less power than that of the conventional system. Further, to improve the capacity the condenser tubes need to be increased.

5.4 COST OF THE SYSTEM

Most of the components used were of the standard type equipment locally available in the market. Frame work, condenser unit, ducting and water circulation system were the other components used. The cost of the system was found to be Rs. 18000.00. Detail of major components alongwith their costs is given in Appendix E.

1.	33.0 86.0	20.5 98.0 290.0	27.0 80.0 45.0	27.0 76.0	30.0 96.0 304.0	1400	290	5.9746	19.4813
2.	31.5 87.0	20.5 94.0 290.5	26.5 80.0 44.0	27.0 78.0	32.0 94.0 300.0	1400	290	5.8368	19.4813
3.	32.0 86.0	18.0 98.0 288.5	24.5 83.0 42.0	24.5 83.0	33.0 95.0 302.0	1375	290	5.3522	19.274
4.	30.0 86.0	19.0 96.0 288.0	25.0 79.0 44.0	24.0 80.0	32.0 94.0 303.0	1375	290	5.3522	18.5345
5.	30.0 92.0	19.0 96.0 289.0	25.5 80.0 43.0	25.0 81.0	31.5 95.0 300.0	1300	290	5.2855	18.1031
6.	31.0 90.0	19.5 96.0 292.0	25.5 80.0 45.0	25.0 81.0	32.0 95.0 302.0	1340	290	5.63007	18.7922
7.	31.0 85.0	18.0 97.0 287.5	25.0 82.0 44.0	24.0 84.0	31.0 95.0 304.0	1300	290	5.3522	18.7922
8.	29.0 78.0	17.0 90.0 286.5	24.0 77.0 45.0	23.0 80.0	29.0 89.0 305.0	1275	290	4.8031	17.8963

No.	Conditions (T_{db}, Ø)	Air entering room (T_{db}, Ø, V)	of air (m^3/s)	Effect (kJ/s)	System (Tons)	COP
1.	33.0 86.0	20.5 98.0 290.0	0.2546	4.265	1.2186	2.524
2.	31.5 82.0	20.5 94.0 290.5	0.2553	4.1915	1.198	2.48
3.	30.0 86.0	19.0 96.0 288.0	0.2563	4.327	1.236	2.72
4.	30.0 92.0	19.0 96.0 289.0	0.2554	4.587	1.3106	2.755
5.	31.0 90.0	19.0 96.0 292.0	0.2557	4.0078	1.145	2.5206
6.	31.0 85.0	18.0 97.0 287.5	0.2577	3.888	1.115	2.385
7.	31.0 85.0	18.0 97.0 287.0	0.2555	4.734	1.352	2.977

5.5 SOUND LEVEL

The present system renders much less sound level inside the conditioned space as compared to the conventional system. The sound levels were measured with the help of Decibelmeter giving reading in decibels . The measurements were taken at a distance of 200 mm and 500 mm away from the grilles for the present system and conventional system. The present system gave 30 db at a distance of 200 mm and zero reading at 500 mm and away from the grille. However, in case of the conventional system the readings were in the range of 50 -69 for the above distances. Hence the present system reduces sound pollution inside the confined space.

CHAPTER - 6

CONCLUSIONS AND SUGGESTIONS

6.1 CONCLUSIONS

The following are the salient features of the present study :

1. The hybrid airconditioner in evaporative cooling mode has good potential for operation in hot-dry climates as well as hot-humid conditions

2. Evaporatively cooled condensing unit has been used in the system thus satisfying dual purpose of evaporative cooling during hot-dry climate and heat rejection from the compressed gas for operation in hot-humid climate.

3. The compressor consumes 1275 to 1400 W depending on the load. This is much below the rating given by the manufacturers. Power consumed by both blowers and pump was found to be 290 W. Thus the power for the system operation in the hot-dry

climate is found to be 290 - 300 W and that in the hot-humid climate 1565 to 1690 W. This gives a saving in power of about 2200 W during hot-dry climates and about 850 W in the hot-humid climate.

4. The system renders much less sound level in the conditioned space as compared to that of the conventional system.

6.2 SUGGESTIONS

1. More number of such models should be tried by varying the size of the condenser coil and size of wood-wool pad to get an optimum result.

2. The hybrid system should be modified to operate as a heat pump such that it may become most efficient year-round airconditioner (i.e. the present system is 2 in one and the suggested system will be 3 in one).

3. The conventional airconditioning system need to be enveloped to get the hybrid modes of operation of the conventional system.

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APPENDIX - A

THEORY OF EVAPORATIVE COOLING

There are two types of evaporative cooling: direct and indirect. The principle underlying the former is conversion of sensible heat of air into latent heat of vaporization of water added directly into the air. In the latter, the air is cooled by the evaporation of water not contacting it, so that the sensible cooling of air occurs without increase in its moisture content [7,20].

The exchange of sensible heat for latent heat under the isolated conditions results in the lowered temperature of air and the air becomes eventually saturated at the water temperature. This process is called 'adiabatic saturation' since there is no external heat exchange betwe the system and surroundings. The practical applications o this principle is found in pump-equipped drip-type coolers, slinger and rotary pad-coolers, most commercial air washer having spray-chamber or capillary design and some textile-mill cooling systems.

In practice however, the water usually gains some external sensible heat in the sump tank, pump and piping. The make-up water entering the sump to replace the evapora and lost water adds heat. Other sources of heat addition include circulatory friction, heat transfer from the surr

saturation in evaporative cooling is merely a close approximation, with considerable evaporation occurring to cool or recool water. When the circulating water is considerably warmer than the air wet-bulb temperature during initial contact, the process resembles that occurring in most cooling towers for cooling warm condenser water. Air and water are jointly cooled and not the air alone. This occurs in direct evaporative cooling systems having 'once-through' or pumpless use of water, as in many small drip-type coolers, some spray-chamber and capillary air-washers. More water is consumed and the cooled air is both warmer and more humid than in true adiabatic saturation.

There are limits to the cooling achieved by adiabatic saturation. The amount of sensible heat removed equals the latent heat required to saturate the air with water vapour. The cooling possibilities of adiabatic saturation vary inversely with the existing degree of humidity of air.

Most commercial direct evaporative coolers deliver air cooled about 70 to 95% towards saturation. This is only 50-70 percent for inexpensive drip-type coolers, but for slinger and air-washer types, this amounts to over 90 percen of perfect humidification. A ratio known as saturating or saturation efficiency (η) is used for rating the performanc of such devices. It is given by:*

$$\eta_{Desert} = \frac{T_{db} - T}{T_{db} \quad T_{wb}} \qquad (1$$

where T_{db} = entering air dry-bulb temperature (C).

T_{wb} = entering air wet-bulb temperature (C).

and T = the dry-bulb temperature of air leaving the system (C).

However, the performance of evaporative coolers will be badly affected if heat enters the process from any source other than the air being cooled. In view of this it is recommended that evaporative coolers be (i) located in the shade wherever possible, (ii) have the coolest possible water supply and (iii) receive the coolest and driest air.

APPENDIX -B

EVALUATION OF CONVECTIVE HEAT TRANSFER CO-EFFICIENTS FOR WALLS AND ROOFS

The values of h_i and h_o were taken from [8] as given below:

For exterior walls:

$$h_o = 0.00391(0.399 + V_w) \text{ kW/m}^2 - C \qquad (B.1)$$

For roof:

$$h_o = 0.00141(1.454 + V_w) \text{ kW/m}^2 - C \qquad (B.2)$$

Where V_w is the outside wind velocity, m/S.

For the inner surfaces of the enclosed space, the convective heat transfer coefficient is found to be the function of the inside air velocity and the temperature difference existing between the room air and inner surface of the wall. In terms of the interior air velocity, V_i and the temperature difference ΔT_i, we get

For interior walls:

$$h_i = C_1 (\Delta T_i)^{0.25} + 0.00391(0.399 + V_i) \text{ kW/m}^2 - C \qquad (B.3)$$

For ceiling:

$$h_i = C_2(\Delta T_i)^{0.25} + 0.00141(1.454 + V_i) \text{ kW/m}^2 - C \qquad (B.4)$$

where C_1 and C_2 are constants, the values of which are given as:

APPENDIX-C

THERMODYNAMIC ANALYSIS OF VAPOUR -COMPRESSION CYCLE BASED ON THE ACTUAL CYCLE

Assuming T_{sup} = 20C, T_c = 5 C; compression, depression and wire drawing equivalant to 4 C and taking

$$T_h = 46\ C,\ T_1 = 10\ C$$

From superheated tables

$$h_1' = 269.5\ kJ/kg = h_1''$$

(assuming $h_1' - h_1''$ to be isenthalpic process)

$$S_1'' = 0.9743 + \frac{(0.9990 - 0.9743)}{275.37 - 268.0} \times (269.5 - 268)$$

$$= 0.097933\ kJ/kg\text{-}k = S_2'$$

(assuming process 1" - 2' to be isentropic)

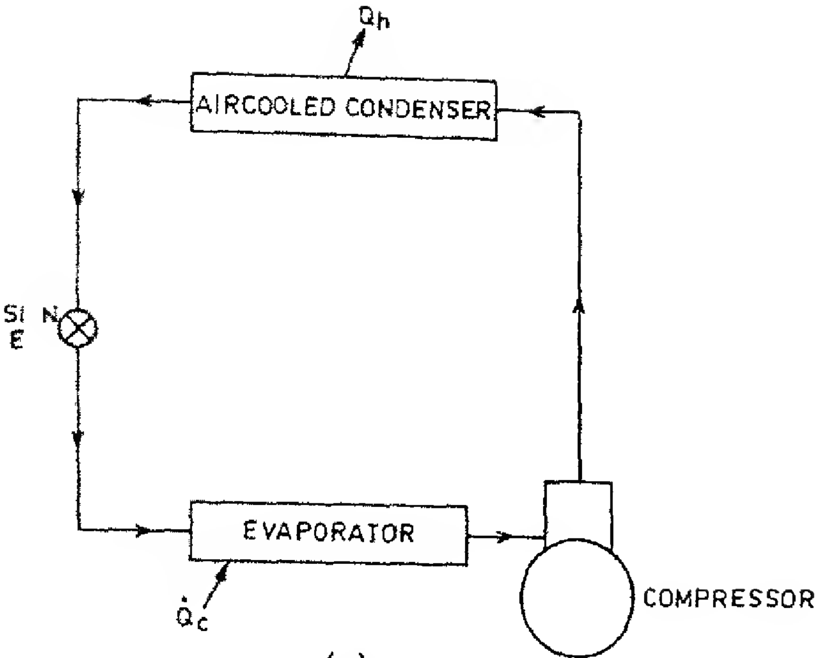
Therefore,

$$h_2' = 300.25 + \frac{309.06 - 300.25}{0.9971 - 0.9731} (0.97933 - 0.9731)$$

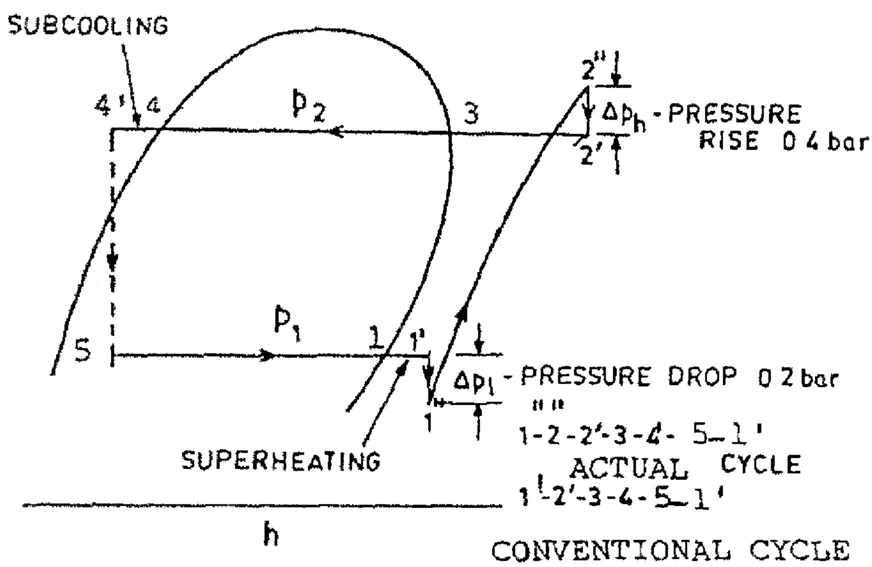
$$= 302.537\ kJ/kg$$

$$V_1'' = 0.043707\ m^3/kg \text{ (from tables)}$$

$$h_4' = 96.67\ kJ/kg$$



(a)



(b)

C.1. Single-stage vapour-compression system

C.2. P-h diagram for single-stage vapour-compression cycle

Pressure ratio $pr = \frac{19.327}{6.02} = 3.210465$

Compressor efficiency for R-22 System is given by

$$\eta_c = \frac{1}{1.13623 + 1.13289 \times 10^{-1}\, pr - 3.34529 \times 10^{-2}\, pr^2 + 4.86757 \times 10^{-3}\, pr^3 - 2.134 \times 10^{-4}\, pr^4}$$

$$= \frac{1}{1.29354} = 0.773073$$

$$h_2'' = \frac{h_2' - h_1''}{\eta_c} + h_1'' = 312.235 \text{ kJ/kg}$$

Considering 1.5 ton system

$$\dot{m} = \frac{1.5 \times 3.5}{h_1' - h_5} = \frac{1.5 \times 3.5}{269.5 - 96.67}$$

$$= 0.0303766 \text{ kg/s}$$

$$\dot{Q}_n = \dot{m}\,(h_2'' - h_4') = 0.0303766\ (312.235 - 96.67)$$

$$= 6.548 \text{ kW}$$

APPENDIX -D

D-1 THERMODYNAMIC ANALYSIS OF VAPOUR-COMPRESSION CYCLE BASED ON EXPERIMENTAL DATA

Evaporator pressure = 5.2855 bar

Condensor pressure = 18.1030 bar

By interpolating the data given in superheated tables we get corresponding Evaporator temperature

$$= \quad 0 + \frac{2}{(5.31 - 4.58)} \times (5.2855 - 4.98)$$

$$= \quad 1.851 \text{ C}$$

and condensor temperature

$$= \quad 46 + \frac{2}{(18.458 - 17.618)} \times (18.103 - 17.618)$$

$$= \quad 47.15 \text{ C}$$

assuming evaporator depression to be 1.85C and compressor depression to be 2.85 C.

with 20 C superheat

Hence $h_2' \quad = \quad 299.32 + \frac{308.01 - 299.32}{1.000265 - 0.97584} \times (0.98454 - 0.97584)$

$= \quad 302.416$ kJ/kg

pressure ratio pr $= \frac{18.1030}{5.2855} = 3.42503$

Compressor efficiency $\eta_c = \frac{1}{1.2980228} = 0.7704$

$\therefore h_2'' = h_1'' + \frac{h_2' - h_1''}{\eta_c} = 266.285 + \frac{302.416 - 266.285}{\eta_c}$

$= 313.184$

Assuming 5C suncooling of condensate

h_4' for 42.15C $= 97.93 + (99.21 - 97.93) \times 0.15$

$= 98.122$ kJ/kg $= h_5$

$\dot{m} = \frac{1.5 \times 3.5}{h_1' - h_5} = \frac{1.5 \times 3.5}{266.285 - 98.122} = 0.03122$ kg/s

Heat rejection $= \dot{m}\ (h_2'' - h_4')$

$= 0.03122\ (313.184 - 98.122)$

$= 6.714$ kJ/s

Power of Compressor $= \dot{m}\ (h_2'' - h_1'')$

$= 1.464$ kW

D-2 CALCULATION OF HEAT REJECTION FROM INLET AND EXIT CONDITIONS OF AIR

a. Inlet conditions of air — Exit condition of air

$T_{db} = 30$ C, $\phi = 92\%$ — $T_{db} = 31.5$C, $\phi = 95\%$

Vapour pressure of air is given by

$$p_v = \phi \; p_s$$

$$= 0.92 \times 0.04246$$

$$= 0.0390632 \text{ bar}$$

Specific humidity $W = \dfrac{0.622 \times p_v}{1.0132 - p_v} = 0.0249424$ kg/kg of dry air

∴ Enthalpy of inlet air $h_1 = 1.004 \, T_{db} + W(2501.4 + 1.88 \, T_{db})$

$$= 380.81 \text{ kJ/kg}$$

b. Exit air condition $T_{db} = 31.5$C, $\phi = 95\%$

$$p_s = \frac{0.04496 + 0.04759}{2} = 0.046275 \text{ bar}$$

$$p_v = \phi \cdot p_s = 0.95 \times 0.046275 = 0.0439612 \text{ bar}$$

Specific humidity $W = \dfrac{0.622 \times p_v}{1.0132 - p_v} = 0.0282117$

$\therefore$ Enthalpy of exit air $h_e = 1.004\, T_{db} + W\,(2501.4 + 1.88\, T_{db})$

$= 392.437$ kJ/kg

c. Mass flow rate of air

Specific volume of exit air $v = \dfrac{287.2 \times T_{db}}{(1.0132 - p_v) \times 10^5}$

$$= \frac{287.2 \times 304.5}{(1.0132 - 0.0439612) \times 10^5}$$

$$= 0.902279 \text{ m}^3/\text{kg}$$

Discharge volume of air $= 300 \times 0.29 \times 0.19$ m^3

$= 16.53$ m^3/min

$\therefore$ mass flow rate $\dot{m}_a = \dfrac{\text{Discharge volume}}{\text{Specific volume}} = \dfrac{16.53}{0.902279}$

$= 0.305338$ kg/s

d. Heat rejection by condenser through exhaust air

$= \dot{m}\,(h_e - h_i)$

$= 0.3053379\,(392.437 - 380.81)$

$= 3.55016$ kJ/s

APPENDIX -E

MAJOR ITEMS USED FOR FABRICATION OF 1.5 TON HYBRID AIRCONDITIONER

SL No (a)	ITEMS (b)	QUANTITY (c)	RATE (Rs) (d)	COST (Rs) (e)	REMARKS (f)
1.	Compressor- SR 1622	1 No	9000.00	9000.00	
2.	Evaporator for 1.5 Ton Airconditioner	1 "	1900.00	1900.00	
3.	Gulmarg Blower	1 "	750.00	750.00	
4.	Evaporator Blower	1 "	600.00	600.00	
5.	Copper Tube				
	(a) 9.525 mm (3/8")	42 m	27.00	1134.00	For condenser unit
	(b) 15.875 mm (5/8")	3 m	44.00	132.00	For water circulation system
	(c) 6.35 mm (1/4")	2 m	20.00	40.00	For condenser unit
6.	Angle Iron				
	(25.4x25.4x3mm)	3.5m	24.00	84.00	For condenser unit
7.	Wiremesh				
	a. 50x50x3mm	1.5m	12.00	18.00	For condenser unit
	b. Cooler type	3.0m	10.00	30.00	For condenser unit
8.	Wood-Wool	1.5kg	10.00	15.00	For condenser unit
9.	Welding gases and copper eutectic	-	150.00	150.00	For condenser unit
10.	GI Tank 900x700x200 mm	1 No	250.00	250.00	For water circulation system
11.	Plastic Pipes	4 m	10.00	40.00	For water circulation system

SL NO	[illegible]	QUANTITY	RATE (Rs)	COST (Rs)	REMARKS
(a)	(b)	(c)	(d)	(e)	(f)
12.	Tullu Vijay Monobloc Pump	1 No.	325.00	325.00	For Water Circulation system
13.	Capillary tube 1740x2.56 mm	1 No.	80.00	80.00	
14.	Aluminium Sheet 2400x1200x1mm	1 No.	250.00	250.00	For ducting
15.	Grilles (for suction and discharge)	2 Nos.	400.00	800.00	
16.	Slotted Angle 55x40x2mm	25 m	30.00	750.00	For frame work
17.	Caster Wheels (100 mm)	4 Nos.	50.00	200.00	For frame work
18.	Valve	1 No.	150.00	150.00	
19.	Capacitors				
	a. Running (36 μFd)	1 No.	200.00	200.00	
	b. Starting (100-120 μFd)	1 No.	62.00	62.00	
20.	Starting Relay	1 No.	70.00	70.00	
21.	Overload protector	1 No.	70.00	70.00	
22.	Miscellaneous				
	a. Fasteners b. Aluminium Rivets c. Electric switches d. Electric Wiring		300.00	300.00	
23.	Fabrication charges			600.00	
			TOTAL:	18000.00	